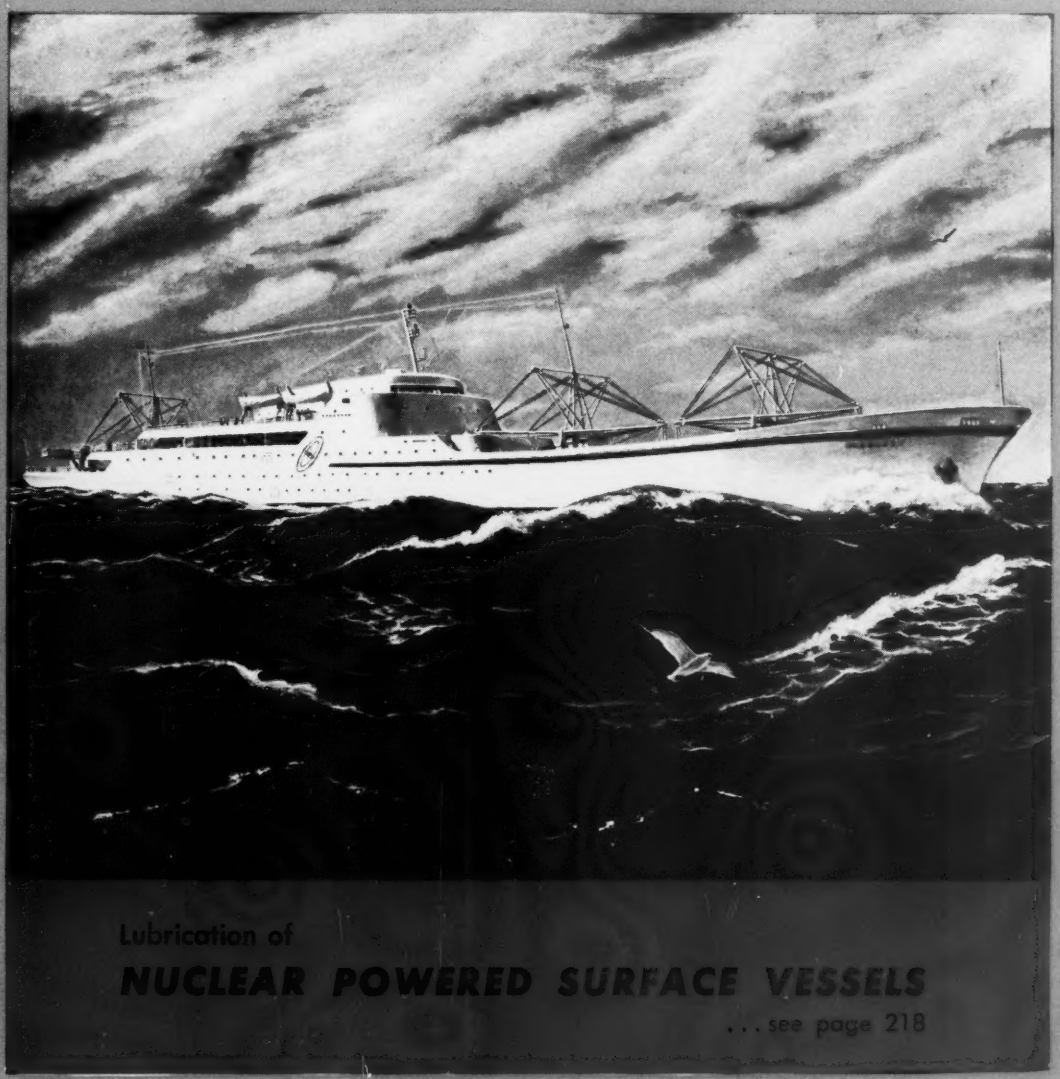
Vol. 17, No. 5 May, 1961

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE



Lubrication of
NUCLEAR POWERED SURFACE VESSELS
... see page 218

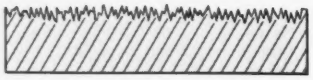
JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

CLIMAX MOLYBDENUM

Molysulfide® News Digest

CLIMAX MOLYBDENUM COMPANY, a division of American Metal Climax, Inc., 1270 Avenue of the Americas, New York 20, N.Y.

MOLYSULFIDE INCREASES TOOL LIFE AS MUCH AS 300 PER CENT



When magnified, a piece of highly polished metal is actually smooth to touch, yet presents a cross-section of saw-tooth irregularity. Molysulfide fills in the valleys, prevents the peaks from tearing, permits plastic deformation of the metal to make a smoother surface.

COMPOUND CONTAINING MOLYSULFIDE REDUCES MACHINERY WEAR-IN TIME

During the wear-in period of machinery, permanent surface damage . . . variously described as *galling, scuffing, scoring, tearing, scratching, excessive abrasion* and *seizing* . . . is an inherent hazard.

After years of experience, leading equipment builders agree that an MoS_2 compound is the best guarantee against wear-in damage. Here are five ways such a compound, applied to moving parts prior to operation, can save you wear-in time, money and possible grief:

1. Shortens wear-in period without use of abrasives.
2. Eliminates stick-slip behavior and resists galling and seizing at bearing pressures far beyond the yield point of any metals.
3. MoS_2 particles have a tenacious adherence to bearing surfaces. Will not scrape off.
4. Need not be burnished into surface. Brush or wipe on a thin coating.
5. Inexpensive in the long run. A little goes a long way.

When writing, refer to CL-103

MoS_2 Provides Greater Resistance to Wear at High Temperatures

The Alpha-Molykote Corporation in Stamford, Conn., is now marketing a new Molysulfide bonded lubricant for ferrous surfaces.

Applied with a conventional paint spray gun, this compound fills the microscopic valleys of the surfaces of tool steel and gives on-the-spot lubrication that allows the chips to glide. Once sprayed, it dries at room temperature to a tough film that has a thickness of 0.0001—0.0008 in. and a coefficient of friction of 0.03 at 100,000 psi.

Extremely resistant to wear at high pressures and temperatures (operating temperature range -300 to 600 F) this new compound requires neither chemical nor mechanical surface pretreatment and, because of its high adherence and low friction coefficient, it increases cutting tool life as much as 300%.

Effectiveness of this lubricant comes mainly from the weak layer-to-layer bonds which allow easy sliding and, within individual layers, from the cohesive forces between Mo and S atoms that are quite strong and prevent penetration by surface asperities.

Minimum coverage coating is 285 sq. ft./lb., and consumption is about 1% of tool weight. Normal lubricating oils, cutting fluids or solvents have no effect on these coatings.

When writing, refer to CL-101

Punch Life Increased from 2,000 to 25,000 Holes in Steel Plate

These punches (11/16") were used to punch 25,000 holes in ½"—1020 cold rolled steel plate. Without a lubricant containing MoS_2, a maximum life of only 2,000 holes could be obtained.

A West Coast manufacturer was experiencing low punch life with the best available lubricants. His multiple punching operation consisted of cold shearing 50 holes simultaneously in a single steel plate. A maximum life of 2,000 holes per punch was obtained with ordinary lubricants.

Then, a compound containing Molysulfide was applied to the punch. The result: punch life was increased from 2,000 to 25,000 holes and, as can well be imagined, considerable savings were effected.

When writing, refer to CL-102

Volume 17, No. 5
May, 1961

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

Contents

LUBRICATION IN PRACTICE

Lubrication Requirements of the Rotary Regenerative Air Preheater ....Joseph Waitkus 226

Hydraulic Filter Requirements of Industrial Servo Systems .....John A. Farris 235

Performance of EP Gear Lubricants in the Rear Axles of an Over-The-Road Truck Fleet —Discussions and Author's Closure .....W. C. Brandow 244

RESEARCH, DESIGN, DEVELOPMENT

The Lubrication Requirement of Nuclear Powered Surface Vessels—Design Considerations .....E. H. Okrent 218

CURRENT NEWS

Letters to the Editor 200

Section News 203

Personals 205

Society News 206

European ASLE Study Trip 211

Report on the 1961 ASLE Annual Meeting 212

LITERATURE SUMMARIZED

Lubrication Abstracts 249

Patent Abstracts 253

TRADE INFORMATION

Advertisers Index 254

Abstracted by Engineering Index and Industrial Arts Index
Cover photo of N S SAVANNAH Courtesy of the New York Shipbuilding Corporation

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LETTERS TO THE EDITOR

Lubrication Engineering welcomes the comments and suggestions of readers with regard to articles published in its pages or concerning topics of interest to those engaged in any phase of lubrication engineering. Letters received cannot be acknowledged, but those felt to be of interest to other readers will be published. The society will not be responsible for the statements or opinions expressed.

Dear Sir:

The following note on a problem of lubrication theory may find some interest among your readers. It concerns the torque required to maintain a pair of concentric cylinders in uniform rotation when they are separated in the annular space, by a viscous fluid. This, apparently simple, problem is not, in fact, as simple as it seems and was incorrectly solved by none less than Sir Isaac Newton (1). To obtain a solution from first principles requires a degree of sophistication and many a beginner has been caught by the subtlety.

To those familiar with the theory of elasticity, and in particular, to the strain relations in two dimensional polar coordinates, the problem has no novelty. But even here if the manipulations are made by purely mathematical means, the difficulty is not brought out in its physical aspects. These physical, or kinematical, aspects are actually treated by Timoshenko (2) with more generality than is required here, and the following developments are merely adapted from this source to suit the fluid problem.

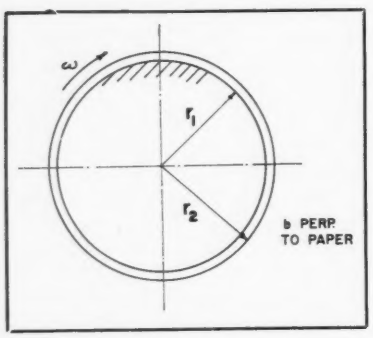

Fig. 1

Referring to Fig. 1: A good, approximate formula for the torque to maintain the outer cylinder in rotation while the inner cylinder is held stationary is formed as follows:

$$T_2 = \frac{\mu\ (2\pi r_2 b)\ (r_2\omega)\ r_2}{(r_2 - r_1)} \qquad [1]$$

μ is the fluid viscosity. The other symbols are definted on the figure.

If, in the same way, the turning couple exerted on the inner, stationary cylinder is written the following result is obtained.

$$T_1 = \frac{\mu\ 2\pi r_1 b)\ (r_2\omega)\ r_1}{(r_2 - r_1)} \qquad [2]$$

It is obvious that some error has been introduced for these torques must be of equal magnitude and they clearly are not.

The correct expression is free of this difficulty and is given by

$$T = \frac{4\pi\mu b r_1^2 r_2^2 \omega}{(r_2^2 - r_1^2)} \qquad [3]$$

Equation [3] can be obtained in the following way and in so doing the difficulty is shown.

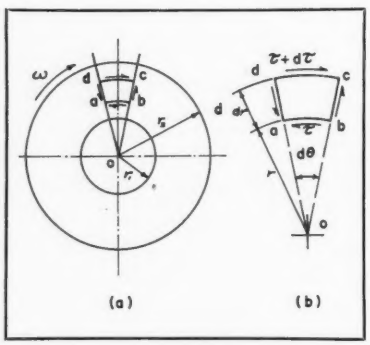

Fig. 2

Referring to Fig. 2; the fluid element of Fig. 2 (a) is shown to an enlarged scale in Fig. 2(b). Since a condition of uniform rotation is being considered the following equation of equilibrium can be written for the fluid element.

$$(T+dT)\ (r+dr)^2 d\theta - Tr^2 d\theta = 0 \qquad [4]$$

Where T is the fluid shear stress at radius r.

Expanding the products and neglecting terms of higher order gives:

$$\frac{dT}{dr} = -2\,\frac{T}{r} \qquad [5]$$

Now, a shear stress in a (Newtonian) fluid can only exist during continued distortion of the fluid elements. Hence, it is now necessary to relate T to the fluid motion. It is here that care must be exercised. If μ is the velocity of a fluid layer at radius r the usual mistake is to say that the fluid distortion rate is given by du/dr. This was the mistake made by Sir Isaac Newton. Because of the curvature of the path the fluid has a rigid body rotation and this, since it does not produce fluid distortion, must be subtracted from du/dr as will now be shown.

If the fluid element abcd of Fig. 2(b) is examined a short time Δt later it will be found to have moved and distorted into a new shape a'b'c'd' shown in Fig. 3.

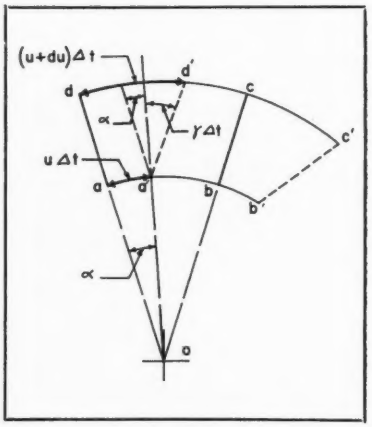

Fig. 3

Let u be the fluid velocity at radius r. The velocity at radius (r+dr) is then (u+du). The point a will have moved in Δt a distance aa' where

$$aa' = u\Delta t$$

The angle α from the figure is given by

$$\alpha = \frac{u\Delta t}{r} \qquad [6]$$

Now, it is clear from the figure that the fluid distortion is the angle $\gamma\Delta t$, where γ is the distortion rate. The angle α is merely the rigid body rotation of the element and does not influence the shear stress.

(*Continued on page* 201)

LETTERS TO THE EDITOR

(*Continued from page* 200)

Again from the figure.

$$\alpha+\gamma\Delta t=\frac{(u+du)\ \Delta t\ -u\Delta t}{dr} \qquad [7]$$

Substituting α from Equation [6] into Equation [7] leads at once to:

$$\gamma\Delta t=\frac{du\ \Delta t}{\ }\ -\frac{u}{r}$$

$$\therefore\gamma=\left(\frac{du}{dr}-\frac{u}{r}\right)$$

$$\text{Hence } T=\mu\left(\frac{du}{dr}-\frac{u}{r}\right) \qquad [8]$$

If T and dT/dr from Equation [8] are placed in Equation [5] the following differential equation is obtained.

$$\frac{r^2d^2u}{dr^2}+\frac{rdu}{dr}-u=0 \qquad [9]$$

The general solution of Equation (9) is

$$u=Ar+\frac{B}{r} \qquad [10]$$

The boundary conditions for the situation shown in Fig. 1 are

$u=0$ for $r=r_1$ and $u=r_2\omega$ for $r=r_2$

These are sufficient for the determination of A and B and lead easily to

$$u=\frac{r_2^2\omega}{(r_2^2\ -r_1^2)}\left(r-\frac{r_1^2}{r}\right) \qquad [11]$$

If u and du/dr from Equation [11] are substituted in Equation [8] the shear stress at any radius is obtained as

$$T=\frac{2\mu r_1^2r_2^2\omega}{(r_2^2-r_1^2)r^2} \qquad [12]$$

The fluid torque due to this shear stress on a fluid layer at r is clearly

$$T=2\pi br^2T \qquad [13]$$

If T from Equation [12] is placed in Equation [13] the desired result is obtained.

$$T=\frac{4\pi\mu br_1^2r_2^2\omega}{(r_2^2\ -r_1^2)} \qquad [14]$$

which is clearly the same at all radii as it should be.

The same result will be obtained if the inner cylinder is turning at ω instead of the outer. If the outer radius is then allowed to become very large, the torque will be essentially that experienced by a cylinder turning in an infinitely large body of viscous fluid. This torque is given by

$$T=4\pi\mu br_1^2\omega \qquad [15]$$

It is interesting to note that this torque is just twice the value which would be obtained if the same device were applied to the approximate formula of Equation [2]. However, in the usual case of relatively thin films, such as those in a bearing, the approximate formulae of Equations [1] and [2] are quite adequate. Moreover, in those cases where the exact formula gives a distinctly different result from the approximate formulae, it is likely that the calculation is invalid because of the possibility of turbulence entering. No doubt a region of speed and clearance exists where the more exact calculation may be worthwhile, e.g., in a rotation viscometer.

The problem concerning the onset of turbulence in this situation is one which has received a good deal of study. Following an investigation of the problem by Lord Rayleigh, Taylor (3) in a classical and difficult paper established the criterion of instability when the inner cylinder rotates. Taylor confirmed his theoretical predictions with some very elegant experiments. His theory shows the motion to be stable at all speeds if the outer cylinder rotates and the inner cylinder is stationary. This was confirmed by experiment up to the highest speed that could be reached. The matter is of importance in the operation of high speed journal bearings. In recent years Wilcock (4) has shown that Taylor's criterion for stability was in agreement with the experimentally observed point at which a sharp change in bearing torque occurred.

Sincerely yours,
F. R. Archibald
Staff Engineer
AC Spark Plug Division
General Motors Corporation

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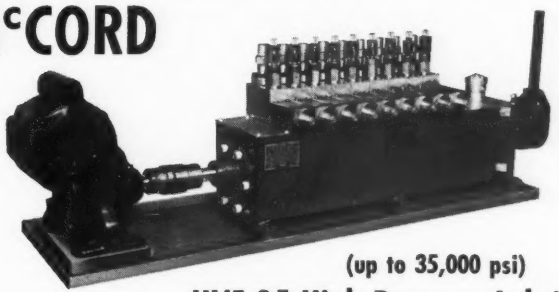

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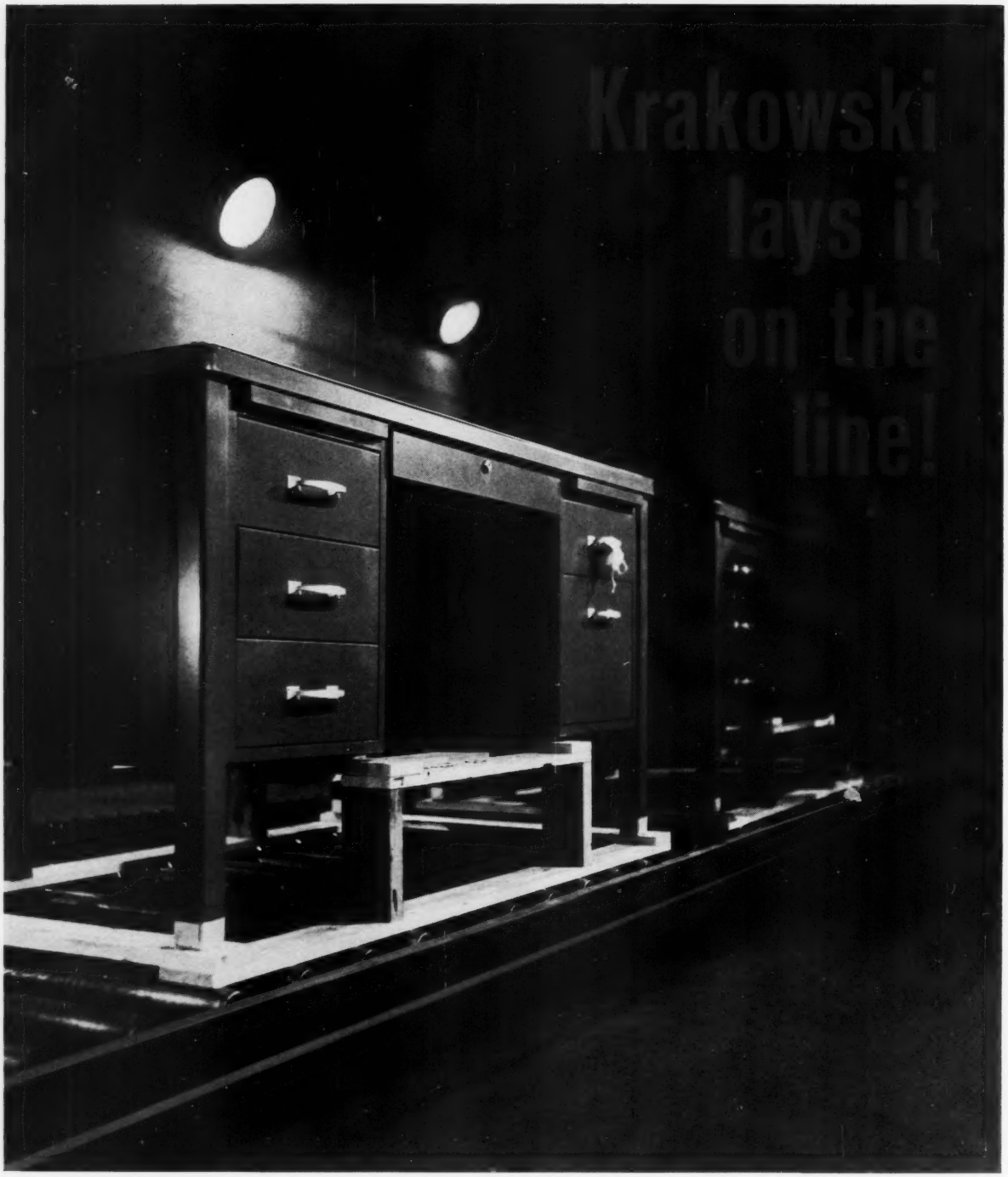

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ASLE President L. O. Witzenburg was on hand at the Rochester Section's March meeting to deliver the main address of the evening. His subject was "Worm Gear Design and Lubrication." He also handled the "What's In It for My Company" approach to ASLE membership. In addition to the large contingent of local section officers and members present, a representative group of the Buffalo Section members traveled through a typical Buffalo-Rochester snow storm to be present. Pictured left to right: Mel Ehrlich, Buffalo Section Chairman; L. O. Witzenburg; Bob Hofmann, Rochester Section Chairman; and Ed Fisher, Rochester Program Chairman.

Saginaw Valley—A representative from Acheson Industries, Inc. will discuss "New Developments in Hi Temperature Lubes" at the May 25 meeting.

Hudson-Mohawk—Mr. E. Brown, General Electric Co., is scheduled to speak at the May 23 meeting. His subject will be "Advances in High Temperature Lubricants."

Cleveland—"Varied Aspects of Plant Lubrication" will be presented at the May 23 meeting by Mr. Charles L. Pope, Lubrication Engineer, Eastman Kodak Co.

Chicago—Guest Night will be observed at the May 18 meeting. Mr. E. J. Krabacher, The Cincinnati Milling Machine Co., will talk on "What's Required and What's New in Cutting Fluids."

The Section members have scheduled their Golf Outing on June 15 at Cog Hill Country Club.

Buffalo—The Annual Outing is planned at the Buffalo Launch Club, Grand Island on June 16.

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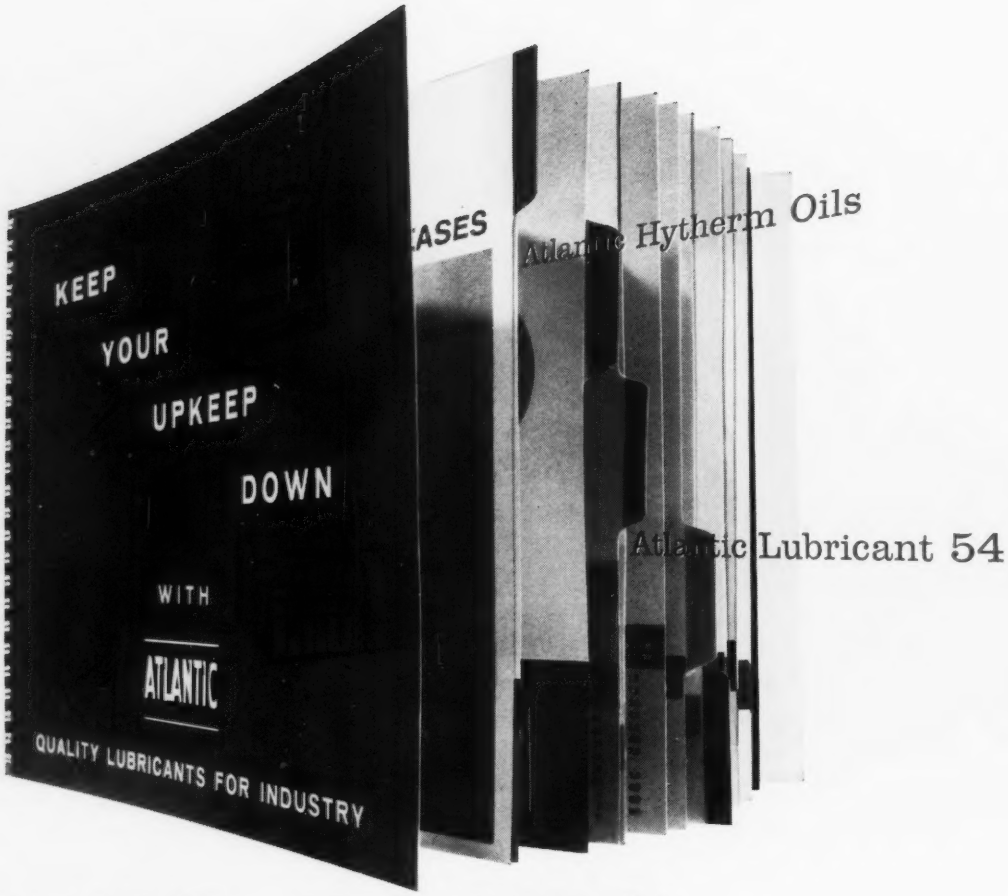

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PERSONALS

Lt. Col. Sidney G. Harris, who left private industry in 1951 to go on active duty with the U.S. Air Force, has retired from military service and rejoined the Joseph Dixon Crucible Co., Jersey City, N.J. He will specialize in marketing development for Dixon's collodial graphite lubricants and coatings. Besides Dixon, he has held executive posts with Petroleum Solvents Corp., McGraw-Hill Publishing Co., and Continental Motors.

LT. COL. S. G. HARRIS

Paul F. Allmendinger has been appointed manager of engineering of the Alemite and Instrument Division of Stewart-Warner Corporation, Chicago, Ill.

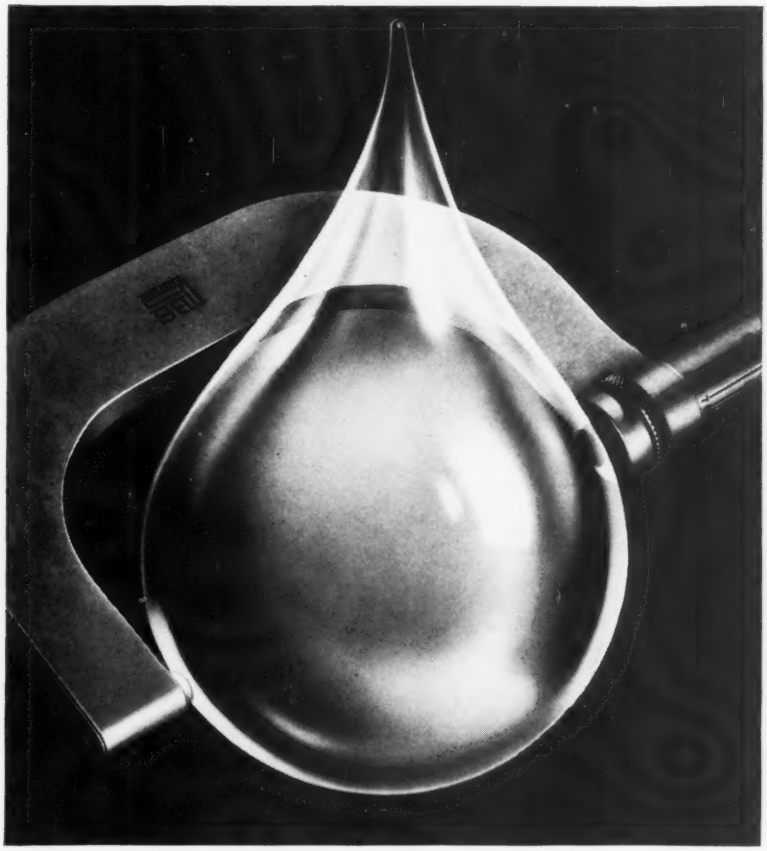
Emerson M. Hough has been named manager of sales technical service for Kerr-McGee Oil Industries, Inc. In this position, Hough will co-ordinate the technical services of the product development and quality control laboratory with the company's marketing division.

Formerly with Sinclair Refining Company for 12 years, he served the last eight years as staff manager of the technical service division.

J. L. Dreher has been promoted from group supervisor to supervising research chemist—Greases and Industrial Lubricants section, Richmond, Calif. Laboratory of California Research Corp., San Francisco, Calif.

(*Continued on page* 254)

How ACCURATE



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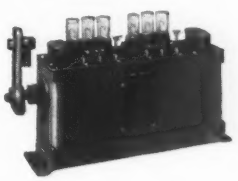

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(*Continued on page* 207)

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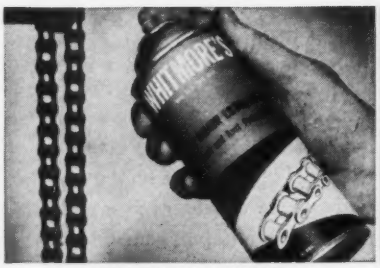

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Penetrates, lubricates, prevents rust. Lubricates deep into chain linkage to reduce friction on pins, rollers, bushings, pressure surfaces. Corrosion inhibitor prevents rust in vital working mechanisms. Lubricates close-tolerance internal surfaces to increase chain life up to 300%. Highly recommended for all climatic conditions to stop rust and corrosion even in salt atmospheres.

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Packaged in 14½ oz. cartridges for handy application with lever-type cartridge guns for constant protection to roller, ball, sleeve bearings and sliding surfaces in "hot" bearing applications or in areas with high ambient heat. Exclusive formulas have no melting or dropping point; moisture resistant; will not wash out; exceptional metal adherence properties; temperature reducing qualities; high extreme pressure values. Available in two grades. Whitmore's Anti-Friction Composition No. 1 (light density), recommended for high speed, high temperature applications. Anti-Friction Composition No. 2 (heavy density), for low-speed high-temperature applications or loosely-fitted bearings. Specify No. when requesting free trial sample.

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SOCIETY NEWS

(*Continued from page* 206)

DETROIT
James T. McCabe
Climax Molybdenum Company
Detroit, Michigan

Earl R. Springstead
Humble Oil & Refining Co.
Detroit, Michigan

Alvin B. Walker
Owens-Illinois Glass Co.
Toledo, Ohio

NORTHERN CALIFORNIA
W. Ehrenreich
Douglas Aircraft Company
El Segundo, California

John H. Smith, Jr.
Columbia Geneva Steel
Pittsburgh, Calif.

BOSTON
Edward F. Colburn
Crocker Burbank Paper Co. Inc.
Fitchburg, Mass.

Thomas C. Smith
Esso Standard Div.
Humble Oil & Refining Co.
Boston, Mass.

ST. LOUIS
Roger E. Hatton
Monsanto Chemical Company
St. Louis, Missouri

KINGSPORT
Donald L. Gray
Oak Ridge, Natl. Lab.
Oak Ridge, Tennessee

SYRACUSE
Earl A. Wittenez
Gulf Oil Corp.
Syracuse, New York

TORONTO
Donald A. Paterson
Monsanto Canada Limited
Toronto, Ontario, Canada

MONTREAL
Ben Segal
Quality Oil Corporation
Montreal, Que., Canada

LOUISVILLE
J. A. Clark
Standard Oil Company
Louisville, Kentucky

CENTRAL OHIO
Robert E. Colliton
The Standard Oil Company (Ohio)
Columbus, Ohio

Thomas B. Janes
Pure Oil Company
Columbus, Ohio

ROCHESTER
Richard C. Smith
J. H. Rae Oil Co. Inc.
Rochester, New York

RICHMOND
Randolph C. Cook
Mobil Oil Company
Richmond, Virginia

HUNTINGTON-CHARLESTON
Bernard Pesetsky
Union Carbide Chemical
So. Charleston, West Va.

OKLAHOMA
Frank D. McGreevy
Champlin Oil & Refining Co.
Enid, Oklahoma

Don A. Amann
California Chemical Co.
Tulsa, Oklahoma

UNAFFILIATED
R. B. Beatson (R)*
Imperial Oil Limited
Winnipeg, Manitoba, Canada

Raymond R. White
Aluminum Co. of Canada
Alcan, Arvida, Que., Canada

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4

GRAFO COLLOIDS *Corporation*
269 Wilkes Place, Sharon, Pa.

Anode casting wheel. Here, molten copper arrives at 2100°F., and is cooled to less than 1500°F. in equipment at right. Pydraul is used in all hydraulic applications close to open flames or extreme heat. It's noncorrosive and fire resistant. Gulf Harmony lubricates wheel rollers. And a Gulf E.P. Lubricant keeps the reduction gears in excellent condition.

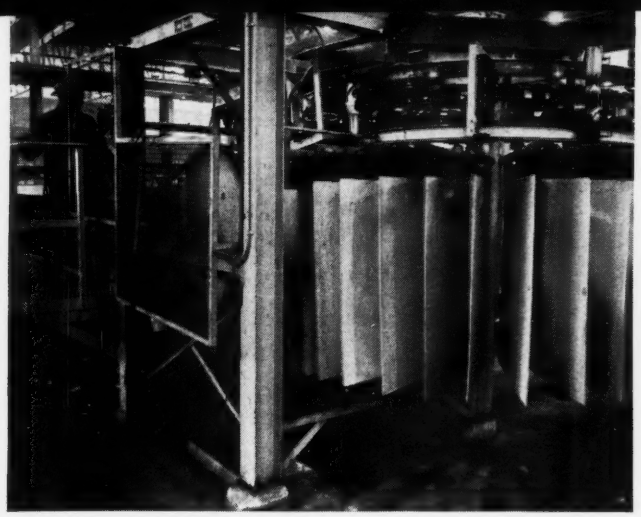
Protects multi-million-dollar facilities with just six Gulf® lubricants...

The Kennecott Refining Corporation, Ann Arundel County, Maryland, refines blister copper. It has invested over 30 million dollars in highly advanced production, quality control and materials handling equipment. Work force: 600 strong. So that's $50,000 worth of equipment per man.

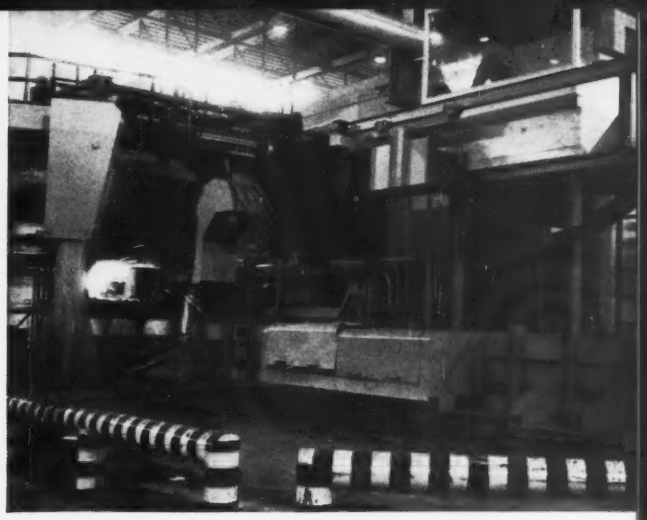
This heavily automated refinery demanded a simple, yet effective lubrication program. To set it up, the Kennecott maintenance staff worked closely with Gulf engineers. The result: a program that requires only a small number of Gulf oils and greases.

For instance, Gulf Harmony® oil is used in hydraulic systems that aren't exposed to extreme heat—assures smooth performance. And Gulf Harmony proves effective in other applications that require an oil with oxidation stability, water separation characteristics and sludge resistance.

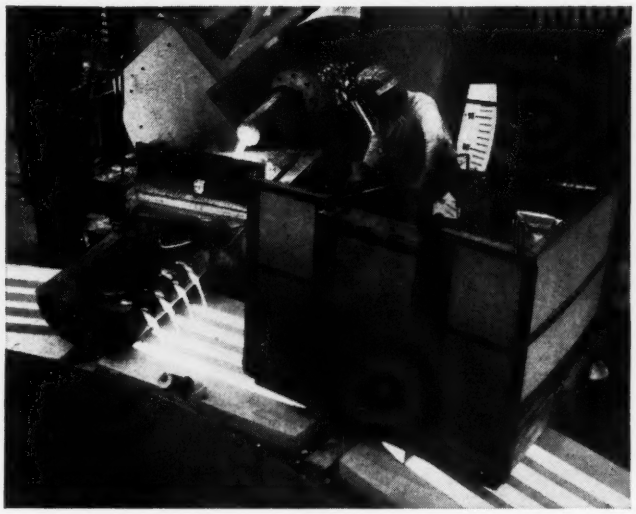
Gulf E.P. Lubricants prevent excessive wear, keep maintenance costs low on equipment that undergoes shock loads and high pressures. Example: reduction gears on anode casting wheel and furnace chargers. A Gulf Sales Engineer reports that "The anode cast-



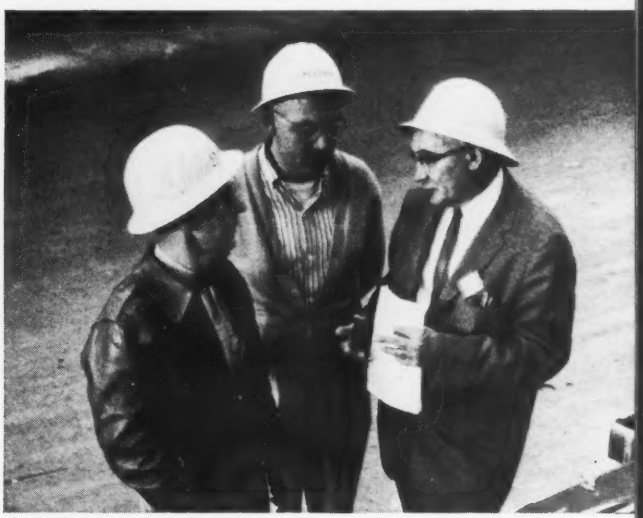
Stripper conveyer. Pulleys and gears are sprayed with Gulf Harmony 76. It's foamless. It's stable. And it provides effective protection against fumes from the electrolytic cells.



Charging machine. It feeds pre-heated cathodes to the 90-ton, 13,500 KVA electric melting furnace. Gulfcrown Grease E.P. is used in the central lube system of all furnace chargers and the rotary pre-heating furnace. It's oxidation-stabilized and protects against rust.



Wire bar casting wheel. The refined molten copper comes from a small induction furnace off the main arc furnace. Gulf Lubcote lubricates the rim gear, drive pinions and guide rolls. And Gulfcrown Grease lubricates the flexible couplings and mold tipping rollers.



Left to right. F. A. Gaidis, Plant Engineer, H. E. Nelson, Machinist Supervisor, and J. E. Foos, Gulf Sales Engineer. A small number of multi-purpose Gulf oils and greases gives the Kennecott Refining Corporation a simple, yet effective lubrication program.

against heat, sludge and wear
GULF MAKES THINGS RUN BETTER!

ing wheel's entire system was recently cleaned, and all reduction gears were in fine shape. No dirt or rust, or any indication of oil breakdown."

Gulfcrown® Grease E.P. is also specified for equipment that meets high pressures and shock loads. Gulf Lubcote® cushions gear-teeth sliding and rolling action on open gears. Reduces noise and wear. Finally, Kennecott uses Pydraul for hydraulic applications near open flames or high temperatures, and uses Gulfcrown Grease in a variety of conventional grease applications.

May we have the opportunity to work with you in setting up a similar money-saving lubrication program with multi-purpose Gulf lubricants? They'll help you cut maintenance costs and simplify lubricant application, storage and handling. Call a Gulf Sales Engineer at your nearest Gulf Office. Or write for free pamphlets.

GULF OIL CORPORATION
Dept. DM, Gulf Building
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"ETHYL" EXTREME PRESSURE ADDITIVES 1 AND 2

Ethyl now offers two unique additives to meet industry's growing need for more effective "extreme pressure" lubricants.

In hypoid gear oils and transaxle fluids, EPA 1 and 2 impart load-carrying capacity for protection against wear under both high-speed and high shock-loading conditions.

In cutting fluids, EPA 1 and 2 perform outstandingly in drilling and tapping titanium and other difficult-to-machine metals—improve surface finish, lengthen tool life.

As high activity additives, both work efficiently in low concentrations. They are composed of mixed chlorinated phosphorus compounds—EPA 1 being relatively high in chlorine by weight; EPA 2 relatively high in phosphorus. Both additives are colorless and free of unpleasant odor. Both permit transparent formulations. Neither is a pro-oxidant.

Complete technical data and samples are available. Write or call for more information.

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ETHYL CORPORATION OF CANADA LIMITED, TORONTO • **ETHYL U.S.A. (EXPORT)** NEW YORK 17, N.Y.

Announcing the
EUROPEAN ASLE STUDY TRIP

Interested in learning the latest European lubrication techniques? Want to know what's happening in lubricant research and development on the Continent? Curious about bearings, centralized systems, greases, or rolling oils used in the steel plants of England and West Germany? Then join the ASLE Study Group. Take the tour that will provide you with the answers first-hand . . . from European Lubrication Engineers.

The American Society of Lubrication Engineers is sponsoring a study trip to Europe in July 1961. This trip has been arranged in cooperation with the Institute of Petroleum in England and the Verein Deutscher Eisenhuttenleute of West Germany. Due to the ASLE's interest in lubricants and steel plant lubrication, the purpose of the trip is two-fold; first, it will afford the opportunity to study the latest trends in European research and development and second, the trip will acquaint men of American industries with the lubrication techniques practiced in European steel making plants.

Two countries—England and Germany—will be visited by the study group. The entire tour will last twenty-seven days including the days of departure and arrival in New York. For those who cannot visit both England and Germany, arrangements can be made to visit either country. The number of personnel that can participate is limited. Membership in the ASLE is not required of participant.

A complete itinerary is as follows:

ENGLAND

July 4—Departure from New York airport.
July 5—Arrive at London airport.
July 6—Visit University Research Laboratory and discuss basic lubrication research with Dr. Bowden.
July 7—Tour Rolay Aeronautical Establishment. High light of the tour—lubrication of the wind tunnel and walking dragline.
July 8—Sightseeing at the individual's leisure.
July 9—Depart for Sheffield by bus.
July 10—Plant tour of Samuel Fox and Company, Ltd., Stocksbridge (near Sheffield).
July 11—Sightseeing tour of Dukeries or the Peak district. Bus departs for Middlesboro in the late afternoon.
July 12—Visit to Dorman Long & Co. at Lackenby. High light of visit—Universal Beam Mill.
July 13—Tour John Sumners & Co., Ltd., Shotton, to view lubrication of modernized hot strip mill.
July 14—Visit Steel Company of Wales at Port Talbot. Departure from London airport.

WEST GERMANY

July 15—Arrive at Dusseldorf airport.
July 16—Sightseeing trip of surrounding areas.
July 17—Meeting with Committee of Lubrication Engineers of the German Iron and Steel Industries at V.D.E. to discuss lubrication techniques in German steelworks. Also visit cold rolling mill and steckel mill of Capito & Klein AG, Dusseldorf-Benrath, and Messrs. Limon de Fluhme & Co., Dusseldorf, manufacturers of centralized grease lubrication equipment.
July 18—Visit the integrated Iron and Steel Works at Rheinhausen. High light of visit will be blast furnace gas turbine, roll bearings, and oil-mist lubrication.
July 19—Tour tube rolling mill facilities of the Phoenix Rheinrorh AG Steel Works at Mulheim. Also visit lubricants testing laboratory at the Technescher Uberwachungsverein in Essen.
July 20—Visit Hoesch AG Westfalenhutte at Dortmund to see new bar mill, continuous billet mill, and strip mill. Also visit Iron and Steel Works at Dortmund-Horder Huttenunion AG.
July 21—Travel to Salzgitter to visit integrated iron and steel works.
July 22—Meet with Committee of Lubrication Engineers of the German Steel Industry in Goslar.
July 23—Sightseeing trip.
July 24—Visit with Professor Vogelpohl at the Frictional Research Division of Max-Planck Institute in Gottingen.
July 25—Visit with manufacturers of anti-friction bearings: Kugelfischer and Kugellager SKF, both located in Schweinfurt.
July 26—Tour of the engineering works at Maschinenfabrik Augsburg-Nurnberg.
July 27—Sightseeing trip to Darmstadt.
July 28—Visit Walzbau—oil circulation equipment—and engineering works of Schenck in Darmstadt. Travel to Koblenz or Dusseldorf.
July 29—Depart at Dusseldorf airport.
July 30—Arrive New York

Since space and time are limited—don't wait. For further information and reservations, contact W. L. Roberts, Applied Research Laboratory, United States Steel Corporation, Monroeville, Pennsylvania.

16th
ASLE ANNUAL MEETING
PHILADELPHIA, PA.
APRIL 11-13, 1961

A look at the forest . . .

More than a thousand of the country's leaders in lubrication research and engineering met in Philadelphia for three days last month. They assembled to work at the business of advancing the state of the art of lubrication engineering. And work they did, in virtually continuous committee sessions, technical presentations and council meetings.

But as the trees so often obscure the forest, the specific problems and actions of the meeting must not be allowed to obscure its broader aspects. Here is a summation of those broader aspects, a look at the forest.

First, the meeting was filled with great promise for the future. Vital needs and practical approaches were clearly defined. It becomes more and more apparent that lubrication is taking its place on a par with such basic considerations as structural materials, configuration, power and power transmission in sophisticated equipment design.

Further, great encouragement was offered to workers in the field of lubrication engineering. New materials and techniques are providing the means for important contributions to industrial efficiency. Recognition will inevitably attend such contribution.

But just as clearly defined as was the promise that lubrication engineering offers for the future, the urgent and immediate need for the Society itself was also apparent. ASLE emerged as a unique instrumentality for preserving the identity of lubrication engineering as a distinct phase of industrial science.

Only through the Society is broad interchange of information concerning the full spectrum of lubrication engineering, from research to application, possible. And only through the Society is general recognition for work in every sector of the field available. Without this full spectrum of interchange and recognition, lubrication engineering could easly breakdown into a group of obscure and unrewarding technical specialties.

In the light of such great promise and urgent need, then, necessary to keep the Society functioning—the work of the directors and committees to maintain policy standards and unit of purpose, the work of the sections to organize meetings and stir interest, the work of individuals to attend and contribute. It is clear that every member of ASLE must look beyond the temptation to concern himself only with some favorite aspect of lubrication technology and to let "George" take care of the society.

Rather, every ASLE member must realize that his society (and his own greatest opportunity for progress and recognition) depends on his own individual interest and participation. . . .

ASLE Board of Directors discuss problems and policy of the coming year.

... In this spirit, start planning now for a year of Society as well as technical activity that will make the decisive decade ahead look like a downhill run when the 1962 ASLE Annual Meeting rolls around—and in this spirit, remember, "St. Louis Next!"

Suppliers of lubricants and lubricating devices reach the heart of the market with exhibits like these at the ASLE Annual Meeting.

And the engineers, fully cognizant of the great service rendered by these suppliers, make a receptive audience. In a way, they come to see and be sold.

The featured social event of the three days of technical sessions and activities was the ASLE Annual Banquet, when the crowd delighted to the excellent talent assembled by the Philadelphia Host Section.

Spontaneous and active audience participation highlighted most of the technical sessions—proof of the real practical value of the material presented.

In committee sessions like these, ASLE cuts across the defined boundaries of the sciences to bring a unique blend of mechanics, physics and chemistry to bear on the needs of industry. Lubrication engineering embraces all technologies and concerns itself with any relative motion problem.

More than 71 papers, covering all phases of lubrication engineering, were presented to eager audiences. Engineers who came with problems went back to their plants with practical solutions. Researchers who came with ideas and theories went back to their labs with new data.

This special three day seminar is a tremendous service to industry. Here, men newly assigned to responsibility for lubrication (or old hands who want a refresher) get the finest course in fundamentals available anywhere—from a panel of lecturers that reads like a who's who in industrial science.

M. E. Dougherty, Acheson Colloids Co., Philadelphia, Pa., Chairman of the ASLE Education Committee.

AWARD RECIPIENTS

Dr. William A. Zisman, Superintendent of the Chemistry Div. of the Naval Research Laboratory, Washington, D.C., received the ASLE National Award and was presented a plaque denoting honorary life membership in ASLE. The manner in which Dr. Zisman has been able to bring fundamental scientific considerations to bear on the solution of practical lubrication problems qualifies him to be named the outstanding man of the year in the field of lubrication.

Each year at the Annual Meeting, awards are presented to deserving members of ASLE. This year, as every year, much time and thought went into the choice of people to receive these special awards. Those considered had to have given exceptionally of themselves and their knowledge in order to further the understanding of all those in the field of lubrication. These men have contributed greatly to the Society, the profession and themselves.

The Walter D. Hodson Award for the best paper written by an ASLE junior author was awarded to J. P. Wettach, Dravo Corp., Pittsburgh, Pa., for his paper entitled "Centralized Recirculating Coolant Systems for Cold Rolling Operations in the Steel and Aluminum Industries."

Robert L. Johnson (left) and Donald H. Buckley, both of the National Aeronautics & Space Administration, received the Captain Alfred E. Hunt Memorial Award for their paper, "The Influence of Silicon Additions on Friction and Wear of Nickel Alloys at Temperatures of 1000F."

Professional recognition and credit for work well done is essential to encourage good men to devote their attention to lubrication problems. ASLE is the sole source for this vital motivation in the field of lubrication engineering. Its awards are eagerly sought and carefully bestowed.

Robert Q. Sharpe, Socony Mobil Oil Co., Inc., New York City, received the Wilbur Deutsch Memorial Award for his paper, "Reducing Leakage from Hydraulic Systems."

A. B. Wilder, Past President ASLE, presented a Certificate of Appreciation to retiring President, L. O. Witzenburg (left).

President Witzenburg presents C. L. Willey, Executive Secretary, with an award for service to the Society. Mr. Willey has accepted a position as Executive Secretary of the Institute of Food Technologists in Chicago.

The Lubrication Requirement of

NUCLEAR POWERED SURFACE VESSELS

—Design Considerations

by E. H. Okrent,

Senior Engineer,
Products Research Division,
Esso Research & Engineering Company
Linden, New Jersey

This paper considers the effect of nuclear surface vessel design on the functional requirements of lubricants. Toward this end, the critical design features of potential nuclear surface propulsion systems have been reviewed to determine the physical requirements imposed on the lubricant by the total environment, radiation being only one of the many factors to be considered. This review has indicated that the lubrication problems of nuclear propelled ships are similar to those of their conventional fossil-fueled counterparts. Radiation stress, the main new environmental feature, is a consideration only in those components which are associated with the nuclear heat source (control rods, etc.), and then it is generally not a controlling consideration. The lubricants for the propulsion gear and its auxiliary systems present no new lubrication problems. However, leakage and primary coolant contamination requirements often take precedence in the selection of the lubricant or lubrication system.

Presented at the 15th ASLE Annual Meeting, Cincinnati, Ohio, April, 1960

INTRODUCTION

The application of nuclear power to commercial power production and ship propulsion has placed a number of new requirements on conventional materials. Of these materials, the lubricants required for proper functioning of the numerous machine elements in the nuclear propulsion system are of prime importance, especially since both the safety and continued operation depend on well-designed lubricants.

This paper presents the results of a study of the lubrication requirements of nuclear powered surface vessels, which was conducted under a contract with the Nuclear Project Office of the U.S. Maritime Administration. This survey was initiated (a) to evaluate the lubrication requirements of nuclear powered surface vessels in the light of present and future designs and (b) to determine if lubricants of enhanced radiation stability will be required for these systems. Part of this study has involved the identification of areas where information is limited and where additional research is needed. This study included a review of both the open and classified literature in the areas of reactor design and radiation resistance of lubricants. In addition, nuclear propulsion system designers were contacted to assess present problems and estimate future needs.*

*No attempt will be made to list the complete bibliography in this paper, however, it may be found in Maritime Administration Report Esso-MA-4. See Reference (1).

An engineering approach was taken in this study, since lubrication cannot be divorced from power or propulsion system design. No discussion of lubrication, be it radiation stability or load-carrying ability, is possible without clearly established design requirements. This is especially true when new or unusual environments are encountered. Radiation stress represents one of these new environmental factors which must be evaluated with the other functional requirements of lubricant stability and "lubricity" in order to develop lubricants with an optimum combination of properties for the given application.

Although this study was primarily aimed at nuclear surface vessel requirements, much of it is directly applicable to stationary nuclear power plants. In fact, from the lubrication standpoint, the surface ship can be considered as a compact version of the stationary plant, but one that presents a somewhat more severe lubrication requirement.

The selection of a nuclear reactor for ship propulsion involves a large number of parameters. As a result, a number of reactor types appear to be applicable. These might include pressurized water, boiling water, organic moderated, gas cooled, liquid-metal cooled, aqueous homogeneous, liquid-metal fueled, fast breeder, pressurized D_2O, and boiling D_2O. With a list this long, a discussion of individual plants, or even general types, would be too detailed for this paper. However, portions of each of these designs will be discussed as it applied to lubrication problems.

COMPARISON OF A CONVENTIONAL STEAM TURBINE SYSTEM WITH A NUCLEAR PROPULSION SYSTEM

Before discussing the lubrication requirements of a nuclear propulsion system, one must first examine ship propulsion systems, both conventional and nuclear, in some detail.

The modern fossil-fueled steam propulsion system, as shown in Fig. 1, is composed of two parts — the heat source and the propulsion gear. The heat source includes such items as the oil-fired boilers, superheater and the blower auxiliary system; the propulsion gear includes the cross-compound turbine, reduction gear and main condenser and feed system. Similarly, Fig. 2 is a schematic of a nuclear propulsion system which indicates that the nuclear propulsion system may also be divided into these same fundamental parts: the nuclear heat source (reactor, steam generator, primary coolant pumps, etc.) and the propulsion gear. As can be seen, the propulsion gear in both the fossil-fueled and the nuclear-fueled systems is essentially the same. The major point of divergence of these two systems is in the heat source. In the case of the fossil-fueled system, the furnace setting acts as a "containment" for the heat source and heat exchange equipment, whereas in the nuclear system, the pressure vessel, the primary and secondary shield, and the reactor containment vessel act as a container for the reactor and the primary coolant loop. From this rather elementary description, it is apparent that the lubrication requirements of nuclear-powered surface vessels resemble the more conventional fossil-fueled system. The only additional features encountered in the nuclear propulsion system are additional environmental factors of radiation stress within the secondary shield, new metallurgies and operation in contact with unusual fluids.

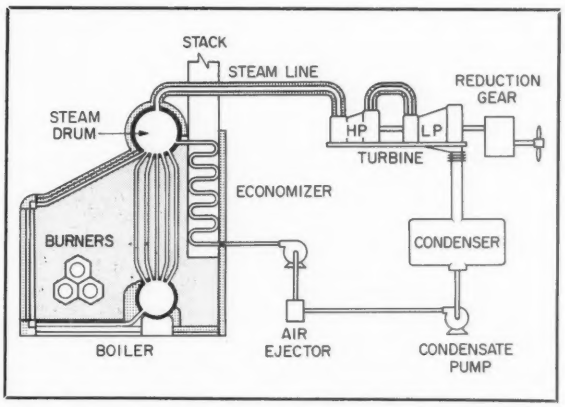

Figure 1—Conventional Steam Generator Power Plant

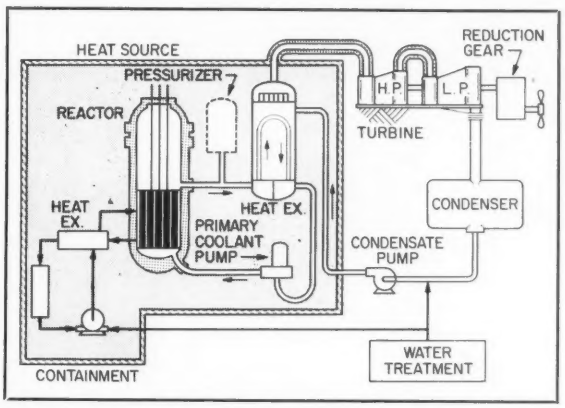

Figure 2—Nuclear Power Plant Dual Fluid Type

LUBRICATION OF COMPONENTS ASSOCIATED WITH THE PROPULSION GEAR

The propulsion gear and auxiliary condensing and feed system of the nuclear propulsion system (turbines, reduction gear, circulation pumps, feed pump, valves) do not present problems from a radiation standpoint.

Most nuclear propulsion systems are based on the concept of containment of the reactor and primary coolant loop. This containment and shielding complicates the problem of energy transfer and has led to a number of heat exchange systems. These can be divided into two basic groups, the dual fluid system (Fig. 2) and the single fluid system (Fig. 3). Two versions of single fluid systems are shown in Fig. 3, the boiling water system (Fig. 3A) and the direct-cycle (Fig. 3B) gas-cooled system.

In the dual fluid system, the working fluid never enters the reactor core and, therefore, has no residual radiation of its own. This is typical of the organic-moderated reactor, the pressurized-water reactor, the gas cooled reactor, the liquid-metal cooled reactor, and

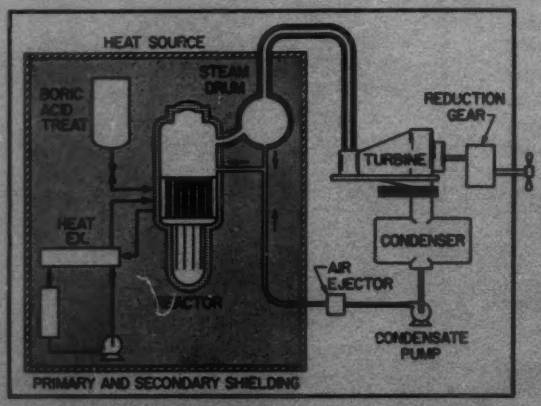

Figure 3A—Nuclear Power Plant Single Fluid Type A. Boiling Water Reactor System

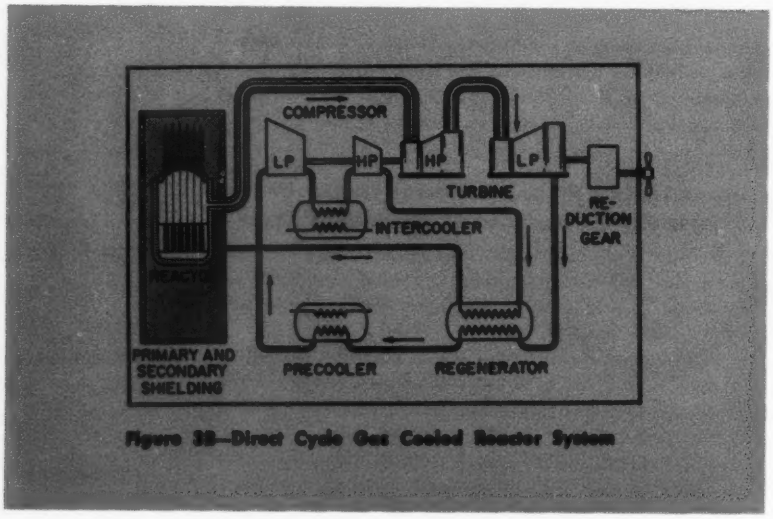

Figure 3B—Direct Cycle Gas Cooled Reactor System

the aqueous homogeneous reactor systems. The only source of radiation flux in the propulsion gear of these systems is that which leaks through the primary and secondary shielding. Since the propulsion machinery is generally accessible to operating personnel, the radiation flux levels should not exceed 0.1-0.5 rads/week. At this radiation level, no evidence of radiation damage will be evident during the useful life of the lubricants.

In single fluid direct-cycle propulsion systems, such as the boiling water and direct-cycle gas-cooled reactors, the problem is somewhat different, since the working fluid will enter the reactor core and will itself be activated. In addition, it can also carry radioactive erosion and corrosion products which generally have long-lived activity.

TABLE 1—Measured Flux in Propulsion Gear and Auxiliary Condensing and Feed System of the Experimental Boiling Water Reactor(1)

COMPONENT	DOSE RATE (mr/hr)
Top of Reactor	1
Sub Reactor Room	10
Steam Dryer	150
Turbine Inlet	50
Turbine Casing	4
Condenser (Above Hot Well)(2)	40
Air Ejector After Cooler	400
Feed Water Filters(2)	100
Ion Exchange Filters(2)	500–1000

(1) VBWR reports similar figures.
(2) Feed pumps are located at this vicinity.

Table 1 summarizes some of the measured radiation flux levels in the propulsion gear and main condensing and feed system of the EBWR(3). The estimated dose rate in the propulsion gear and machine space of a proposed 22,000 SHP, boiling-water tanker is shown in Table 2(4).

In the case of the direct-cycle gas-cooled system designs, dose rate information is not as extensive since these designs have not reached the same state of development as the boiling water systems. However, in the helium-cooled MGCR system, the estimated dose rate at the turbine will be 0.6 rads/hour (10^5 rads/20 years) if the helium is maintained at a purity of 0.1 ppm Ar^{40}. In systems comparable to the marine gas-cooled reactor but using other coolants, the dose rates would be somewhat higher.

Radiation dose rates in the propulsion gear and auxiliary condensing and feed system are summarized in Table 3. These data were obtained from discussions with designers, estimates from existing plants (stationary, surface and submersible) and from nuclear ship proposals; and in some instances these will change as designs are changed. This table shows that the dose rates in the propulsion gear of nuclear-powered surface vessels are quite low, even in the direct-cycle gas-cooled and the direct-cycle boiling water systems. This point is emphasized more clearly when one compares the calculated and measured radiation dose rates with the 25% damage tolerance dose for conventional lubricants in low intensity fields. The first evidence of radiation damage to these materials would not appear for at least 30 years even in the most critical applications. Thus, it may be concluded that the lubrication of the propulsion and auxiliary condensing and feed system components does not present a radiation stability problem. The deciding factor in lubricant selection will be such conventional factors as oxidation stability and load-carrying ability. Oxidation stability will be especially important to the long-term retention of lubricant properties since turbine lubricants operate between 140 F and 200 F under mildly oxidative conditions.

TABLE 2—Estimated Dose Rate in the Propulsion Gear of 22,000 shp Boiling Water Tanker

UNCONTROLLED AREAS	DOSE RATE (mr/hr)
Machinery Passageway (Forward of Machinery)	0.07
CONTROLLED AREAS	
Turbogenerator	39
Turbogenerator Cross-Over Pipe	216
Demineralizer(1)	10

(1) 15 minutes after shutdown, all others at full power.

TABLE 3—Estimated Radiation Dose Rates in the Propulsion Gear of Nuclear Powered Surface Vessels (Rads/Week)

COMPONENT	BWR	PWR	OMR	MGCR	LMC
Turbines	8-16	0.1	0.5	100	Negligible
Cross-Over	34-50	0.1	—	—	Negligible
Machine Space	0.012	0.1	0.5	25	Negligible
Condenser	8-16	0.1	—	—	Negligible

Allowable Radiation Dose

People:	0.15 Rads/Week
Lubricants:	
0% Damage	10^7 Rads
25% Damage(1)	10^8 Rads

(1) Dosage at which 25% change in critical physical property has occurred, i.e., viscosity, interfacial tension, oxidation life, etc.

BWR—Boiling Water Reactor
OMR—Organic Moderated Reactor
LMC—Liquid Metal Cooled
PWR—Pressurized Water Reactor
MGCR—Marine Gas Cooled Reactor

LUBRICATION OF COMPONENTS ASSOCIATED WITH THE NUCLEAR HEAT SOURCE

The nuclear heat source and its associated control and heat-transfer equipment are located within the reactor compartment. Some of the components that must be lubricated are within the secondary shield system, where radiation stress is a consideration. These components include pumps and compressors, control-rod drives, valves and valve actuators, motor auxiliaries and fuel-handling equipment. The following paragraphs will discuss the criteria for lubricant selection in these components.

Pumps and Compressors

The first of these necessary heat-source auxiliaries are the primary coolant pumps and compressors. This group includes all of the pumps associated with the heat source (primary coolant, auxiliary, feed, etc.). The types of pumps available for these services are varied, making a discussion of specific requirements impractical. However, the basic lubrication requirements are identical with similar equipment in conventional power and propulsion systems. In addition to these conventional requirements, the nuclear system presents a number of unique requirements which must be considered.

Considering the first unique feature of the nuclear environment, radiation stress, the survey revealed that the radiation dose rates in these lubricated components are well below the limiting capability of conventional lubricants. This is particularly evident when one compares the maximum expected radiation dose rates in these components (Table 4) with the radiation resistance of conventional lubricants. Table 4 indicates that radiation stress is a consideration but not a limiting one. In these systems, conventional lubricant requirements will still be controlling.

TABLE 4—Estimated Maximum Radiation Dose Rate in Pumps and Compressors Associated with the Nuclear Heat Source (Rads/Hour)

Reactor Type	Primary Coolant Pumps or Compressor	Charge and Emergency Cooling Pumps	Purification Pumps
Pressurized Water Reactor	10^2-10^3	60-300	1-5
Boiling Water Reactor	10^2-10^3	60-300	1-5
Organic Moderated Reactor	Negligible	Negligible	Negligible
Gas Cooled Direct Cycle	1-10	Low	Low
Liquid Metal Cooled	10^4-10^5	—	—

Lubricant Radiation Stability

0% Damage	10^7 Rads
25% Damage(1)	10^8 Rads

(1) Dosage at which a 25% change in a critical physical property has occurred, i.e., viscosity, dropping point, oxidation life, etc.

In the liquid-metal cooled system, the maximum radiation intensity could place a strain on existing products. However, conventional lubricants cannot be considered for this service, because the operating temperature approaches 700-800 F, well beyond the capability of conventional lubricants. In this system, electromagnetic pumps and canned rotor pumps have found application. However, freeze-seal type pumps have been developed, which have shafts that extend through the shielding, so that the pump motor and bearings are outside the radiation flux. With this design, petroleum lubricants can be used to cool the liquid-metal shaft seals and lubricate the drive motors.

Since radiation stress is not a limiting consideration, the question that naturally follows is how extensively are petroleum-based lubricants used in these primary coolant pumps? This question is answered in Table 5, which summarizes primary-coolant pump lubrication in a number of nuclear power and propulsion systems being considered for surface-ship propulsion. This table indicates that organic lubricants are used only when primary coolant loop contamination is not a problem (or when it is provided for, as in the MGCR) or when zero primary-coolant leakage requirements permit. Thus, leakage and contamination requirements are apparently major criteria in pump lubricant selection.

For example, in the pressurized-water and boiling-water reactor systems, where radiation stress does not limit the use of petroleum lubricants, high-cost water-lubricated (primary fluid) canned rotor pumps have been used. This type of pump was selected primarily

TABLE 5—Primary Coolant Pumps for Nuclear Reactors

Reactor Type	Primary Coolant Pump Types	Lubricant Type	
		Drive Motor	Pump Bearings
Pressurized Water	Canned Motor Centrifugal	Water Cooled and Lubricated	
Boiling Water	Limited Leakage Centrifugal	Grease-Oil(1)	Water(2)
Organic Moderated	Standard Chemical Process Pumps	Grease-Oil(1)	Lube Oil, Grease or OM
Gas-Cooled-Steam(4)	Blowers	Grease-Oil(1)	Lubricating Oil
	Canned Compressors	Gas Lubricated	
	Turbo-Compressors	Turbine Oils	
Gas-Cooled-Direct Cycle(4)	Turbo-Compressors	Turbine Oils	
Liquid Metal Cooled	Sump Type Propeller Pump(3) (external shielded motor)	Grease	Liquid Metal
Liquid Metal Fueled-Steam, Gas, or Mercury Vapor Cycle	Submerged Centrifugal	Liquid Metal	
Fast Breeder	Canned Rotor Pump	Liquid Metal	
	Electromagnetic	N/A	
Aqueous Homogeneous	Canned Rotor Centrifugal	Water	

(1) Depends on speed and load, generally grease is used in sealed-for-life systems.
(2) Conventional lubes may be used with proper seal arrangements.
(3) Motor located outside shielding for ease of maintenance, pump operates in liquid metal.
(4) Located outside shielding.

to avoid leakage of the primary fluid from the system and prevent the contamination of the primary coolant with lubricant which might leak in through the primary coolant pump seals. The designers believe that lubricant leakage could foul the heat transfer surfaces and cause subsequent failure of the fuel elements. No published experimental data have been found which define the allowable lubricant leakage requirement for water-cooled and moderated-reactor systems.

Lubricant leakage and contamination requirements are also important in the gas-cooled reactor systems. Here, petroleum oils are used to lubricate the primary coolant compressors or blowers and, in the direct-cycle system, the power turbine. Leakage of lubricant past the oil seals into the gas stream and leakage of primary coolant (gas) into the turbine or compressor lubricants are important problems. There are a number of proposed solutions which appear to be promising; however, all use a lubricant clean-up system of some form. The clean-up system as envisioned requires that the bulk of the coolant gas (helium in MGCR) be removed from the oil stream and no more than 2 ppm of lubricating oil be retained in the primary coolant gas. In order to design such a system, several questions need to be answered. These are:

1. Solubility of helium (or other gas) in turbine oils at temperatures up to 200 F and pressures near 600 psig.
2. Effect of dissolved gas and pressure on the lubricant viscosity.
3. Foam and entrainment effects.
4. Vapor pressure of the lubricant and evaporation rate.

These four points are being investigated in detail in these laboratories and the detailed results will be published elsewhere. However, some experimental work indicates that for the first three items no serious problem exists with helium. The vapor pressure problem is not so easily solved, since the 2 ppm lubricant contamination requirement limits the vapor pressure which can be tolerated in turbine oils for this application. Many turbine oils containing low-molecular-weight oxidation inhibitors will not meet this requirement, even though their base oil has the desired vapor pressure. In these systems, high-molecular-weight oxidation inhibitors will be required, and pre-stripping of light components may be necessary to meet the 2 ppm requirement.

This limiting concentration of lubricant permitted in the primary coolant again is based upon very limited information. The author has been unable to find any reference in the published literature to definitive experimental work aimed at establishing this requirement. The only available reference is some information obtained by Fraas (ORNL) (5) during a visit to Calder Hall which indicated that lubricant contamination in excess of 7 ppm led to fouling of heat exchange surfaces, while at 4 ppm no fouling was observed.

Control Rod Drives

A second moving part associated with the nuclear heat source is the control-rod drive mechanism. These mechanisms are required to move rods composed of fissionable, reflecting, or neutron-absorbing material to control the reactivity of the nuclear heat source. These rods must be positioned accurately, maintain predetermined fixed rates of motion, and must be capable of rapid motion in the event of safety shutdown. Thus, the insurance of the continued operation of these drives is of major importance to both the safety and control of the nuclear heat source.

The control rod drive mechanisms for stationary and surface propulsion systems include rack-and-pinion gears, lead screws, hydraulic or pneumatic systems, magnetic jacks, and winch-and-cable arrangements. Generally these mechanisms may be classed in three groups: (a) those which operated totally immersed in the primary coolant, (b) those which are only partially immersed, and (c) those which operate in gas.

The first category is exemplified by the Shippingport reactor where each control rod drive is moved by an electric motor, the armature of which is a rotor nut, through which a lead screw runs. The motion of the armature raises or lowers the control rod. The rotor nut, lead screw and support bearings for this unit are located inside a pressure tube attached to the reactor pressure vessel. The motor stator and position indicator are located outside the pressure tube. The armature is supported on ball bearings of stellite balls and races which are lubricated by the primary coolant fluid. Performance of these systems indicates 500,000 cycles of operation without excessive wear.

The second group typified by the control rod drives of the APPR, EBWR, ALPR and the NS Savannah uses a seal system to isolate the primary coolant from all or a portion of the drive mechanism. Primary coolant is used to lubricate only a part of the system; most of the lubrication requirements are handled by conventional lubricating greases and oils. For example, the primary coolant is used to lubricate the rack, pinion and rollers of the APPR design, while conventional greases and oils are required for the drive motors, bearings and gear box. In addition, some of these drives require hydraulic scramming when operating in a pressurized water system. Organic hydraulic fluids can also be used for this purpose, since the water and steel above the core provide adequate shielding.

The third group of systems, the "gas-immersed systems" are troublesome from a lubrication standpoint, since the lubricants are in contact with high-temperature gases (150-200 C). The principal lubricant requirements for this application are high-temperature stability and non-reactivity with the coolant. Radiation stability is not a problem. Greases are generally used in these systems and the major quality considerations are low oil-bleeding, good thermal stability, structure stability, and compatability with reactor construction materials. When CO_2 is the coolant gas, conventional soap-thickened greases should not be used, since the soap structure is destroyed by reaction with CO_2. Non-soap-thickened greases are recommended for these applications.

There is a considerable spread of estimated dose rates for the control-rod drive mechanisms in the literature for the various reactors. This spread is due in part to the design differences. However, one additional point evident from the literature is that design

dose rates in the region of operating parts are very rarely defined. Therefore, one must resort to estimating the dose rates based upon dose rates in adjacent areas. Table 6 presents the maximum range of dose rates estimated from the literature and from discussions design personnel.

Considering the merchant ship proposals, the situation is very much better, since the dose rate data, even at this early design stage, is better defined. The only maritime reactor system where dose rate data were not available in the literature was the MGCR. Table 7 summarizes the dose rate data for the maritime reactor proposals.

An examination of Tables 6 and 7 shows that the radiation dose rates are not excessive for these applications. At these dose rates, it would take 10-20 years before exceeding the 10^8 rads total dose tolerated by conventional lubricants.

TABLE 6—Estimated Maximum Dose Rates in Control Rod Drive Mechanisms(1) Rads/Hour

Pressurized Water	$10\text{-}10^3$
Boiling Water	10^2
Organic Moderated	$10^2\text{-}10^4$
Gas Cooled	? 0.1-10(2)

(1) Based on stationary power plants, and submersible and surface ship systems.
(2) Based on British gas-cooled experience.

TABLE 7—Estimated Dose Rates in the Control Rod Drives of Maritime Reactors—Rads/Hour, Full Power

Pressurized Water (NS Savannah)	10-20
Boiling Water (22,000 SHP Tanker Proposal)	0.250
Organic Moderated (38,000 DWT Tanker Proposal)	2.2×10^2
Gas Cooled System (MGCR)	?

Control and Stop Valves—Motor Auxiliaries

The control and stop valves and their motor auxiliaries, which are located within the reactor containment, are exposed to a radiation flux which varies from 10-300 rads/hour. The type of valve selected and its actuation system depend entirely on the control required and the physical layout of the plant. These valves may use hydraulic, air, or electric actuation and may require sliding or rotating motion, etc. However, the primary coolant valves of maritime reactors generally are stop valves, and operate either full-open or full-closed, so that continuous throttling action is not required. The present submersible power plants utilize hydraulically operated valves which use water as the hydraulic fluid.

Hydraulic-actuated valves have been adopted because of the early difficulty involved in incorporating air operated or electric motor driven valves in hermetically sealed systems which was required to meet the zero leakage requirement. These valves are lubricated by the primary coolant. Significant savings can be obtained in plant costs if electric or air operated valves of sufficient reliability and leak tightness can be developed. In the NS Savannah design, the primary coolant valves are electric motor driven, geared, limit-torque valves. These valves will be grease-lubricated and operate at an ambient temperature of 140-150 F, in semi-sealed gear boxes, and must withstand transient temperatures up to 320 F. Conventional high-temperature greases should give satisfactory service in this application.

The motor auxiliaries and valve operators generally use grease-lubricated sealed-for-life bearings. With a 20-year service life expectancy for these motors (150 rads/hour at full reactor power) petroleum-based greases would just begin to show signs of observable radiation degradation at the end of this service life (10^7 rads). Therefore, it appears that conventional petroleum-based, soap-thickened greases will be satisfactory in this service.

Fuel Handling Equipment

The lubrication and design of fuel handling equipment depends upon the duty cycle imposed by the reactor's operating characteristics. For example, long periods between loadings of surface vessels allow handling systems to be portable with all other reactor considerations taking precedence. Such fuel-handling devices can operate slowly, have a relatively long setup time and can be moved manually with some mechanical assistance. Furthermore, it can utilize materials with a limited life span. This type of system needs to have a high degree of reliability during operation, but long periods between loadings afford accessibility for maintenance. In this refueling procedure, the reactor is shut down, a portion of the reactor pressure vessel is removed and the refueling is conducted with water or auxiliary shielding present. The radiation dose rates in these systems are quite low—180 mr/hr to 4 rads/hr —although higher dose rates exist in the shield-cask (up to 10^5 rads/hr). Generally, the parts requiring lubrication do not see the higher dose rates. In addition, the components requiring lubrication (servo motors, hoists, gears and latching mechanisms) are available for relubrication. Conventional lubricants will again be more than adequate.

Short periods between loadings impose a different set of requirements. The handling system must be a permanent, integrated portion of the nuclear heat source. Loading and unloading must be efficient; maneuvering must be fast and accurate. Long life is desirable but maintenance is possible. Within this group it is necessary to sub-divide this type into two categories: (a) intermittent refueling with the reactor shut down and (b) continuous refueling with the reactor at full power.

In the first category are reactors such as Shippingport, SRE, and Enrico Fermi reactor, where radiation dose rates may be high for short periods of time during fuel transfer, but relubrication is possible after use. Lubricants of enhanced radiation stability will not generally be required for this application because reliability during operation is the prime requisite of this equipment. Relubrication of parts exposed to high radiation stress with a low-cost conventional grease is a cheap price to pay for satisfactory operation. The second case which is typical of the U.K. gas-cooled systems and some U.S. gas cooled designs requires continuous rotation of fuel elements. The British designers feel that they require a lubricant which will withstand 10^9 rads total dose during a six-month period in this application. The dose rates and total dose requirements of the U.S. air-cooled reactor fuel systems are not available in the literature at this writing.

INFLUENCE OF FUTURE REACTOR SYSTEMS ON LUBRICANT REQUIREMENTS

Most of the previous discussion has centered around such reactor systems as the pressurized water, the boiling water, OMR and gas-cooled systems. The lubrication requirements of the newer reactor concepts (aqueous homogeneous, liquid-metal fueled, fused fluoride, breeders, etc.) are somewhat speculative since only prototypes have been designed to date. Detailed information on surface propulsion systems based upon these reactors is non-existent. Therefore, the question of what direction future nuclear surface propulsion systems will take and what effect these design options will have on lubricant requirements is difficult to answer.

However, the various options open to the designer can be discussed in a general way to see how lubricant requirements will be affected. The options open to the nuclear propulsion system designer include: (a) coolant type, (b) physical state of moderator and fuel, (c) distribution and location of fuel and (d) fuel cycle. A discussion of each of these options, and its effect on lubricant requirements would not be a fruitful approach, since no matter which option the designer takes, the ultimate result is the same—that is, the lubrication requirements would not be markedly altered, provided reactors are designed with today's ground rules.

The major reason for the previous statement (that radiation resistance requirement will remain unchanged as long as present ground rules apply) is based primarily on shielding considerations. As long as present ground rules apply) is based primarily on shielding considerations. As long as manned surface propulsion systems have reactor compartments which are accessible fifteen minutes after reactor shutdown, large amounts of shielding will be required. The most economical shield design is obtained when the bulk of the shielding required for equipment and personnel protection is placed as close to the reactor as possible. Thus, with the exception of those systems circulating large quantities of radioactive fuels in the primary coolant systems, future propulsion systems designed under present ground rules would not present any increased radiation resistance problem.

In the case of the circulating fuel reactors (which are all of the dual fluid type), some increase in the radiation stress at the primary coolant pump over current designs is anticipated. However, the prototype systems generally either provide adequate shielding when organic lubricants are used or use pumps which require no organic lubricants. The possibility of replacing the system with pumps which require organic lubricants is remote since these systems have far more stringent leakage and contamination requirement than the BWR or PWR systems.

The picture presented of the future lubricant requirements is critically dependent on retaining the current design ground rules. These ground rules could change, in which case future designs could be more critical of radiation stability in lubricants. For example, one could consider a completely submerged, remotely controlled nuclear propulsion system used either as a submarine tug or as a power plant for a hydroplane ship. In such a system, it might be economical to use a minimum of shielding around the reactor, house the nuclear heat source and propulsion gear in the same compartment, and use the steel and water required to sink the system as shielding. The radiation stresses in the lubricants of such a hypothetical system would be enormous, and special fluids would be needed.

Thus, it is apparent that a change in any or all of the present ground rules can markedly shift the lubricant requirements. This means reactor design concepts must be continually reviewed and examined to insure the availability of proper lubricants for future reactor types as needs arise and the design ground rules shift.

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Rotary regenerative air preheaters are synonymous with waste heat recovery. To assure the highest degree of reliability and availability, lubrication is considered with more than ordinary care when engineering various components of the equipment. This paper describes the regenerative principle of heat exchange, illustrates several designs for a wide range of conditions, and directs special emphasis to several components where lubrication is required.

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Lubrication Requirements of the Rotary Regenerative Air Preheater

INTRODUCTION

Undoubtedly, the term "rotary regenerative air preheater" will be completely foreign to most lubrication engineers. This lack of familiarity is understandable since the function of the equipment, and the position it occupies in the plant arrangement, is remote from areas of common knowledge and inspection. For this reason, a brief introduction might be of some value.

Today, waste heat recovery is assuming a new level of importance in the economics of many processes and plants. This is due to the continuous rise in fuel and labor costs. Exhaust gases, from boilers and furnaces, or any apparatus serving to apply heat from natural or residual fuels to some useful purpose in a production process, are a large source of waste heat. Returning this rejected heat to the system often results in attractive savings and improvements of many types.

The rotary regenerative air preheater is a waste heat recovery device. It extracts heat from exhaust gases and transfers it to ordinary air. In the exchange, the air is elevated in temperature. When the heated air is mixed with fuel at the burner it can serve to reduce fuel consumption or, when returned to the process, it can serve to improve the product and/or process in some form.

Presented at the 15th ASLE Annual Meeting in Cincinnati, Ohio, April, 1960.

Figs. 1, 2, 3 and 4 illustrate several forms of the rotary regenerative air preheater. Type "V" is the vertical design and type "H" is the horizontal design. The terms—vertical and horizontal—allude to the direction of flow of gases and air through the equipment.

The basic principle of operation is illustrated in Fig. 5. Gases and air flow in opposite or counterflow direction while a continuously rotating element cuts through both streams. This rotating element is a mass of thin metallic plates, fabricated to special dimensions and spaced to permit the free-in and outflow of gases and air. As the element passes through the hot gas stream, heat is absorbed which is carried into the air stream. The air, flowing over the hot plates, absorbs the heat and is elevated in temperature to some predictable level.

The principle is simple and highly efficient. The regenerative principle of heat exchange is considered to be the most efficient available. This accounts for its wide acceptance in waste heat recovery, particularly in the electric utility industry where better than 85% of the steam generating capacity is supplemented with heat recovery equipment of the rotary regenerative type.

This equipment with its rotating or moving components naturally falls into the classification of a machine. It is subjected to lubrication considerations for the same general reasons. To engineers unfamiliar with the equipment, the lubrication requirements may appear no different than those encountered with other

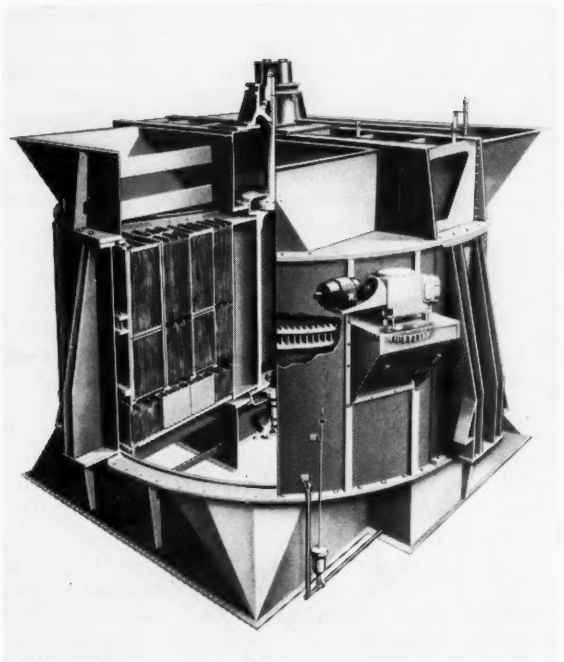

Fig. 1. Typical "V" Type Rotary Regenerative Air Preheater

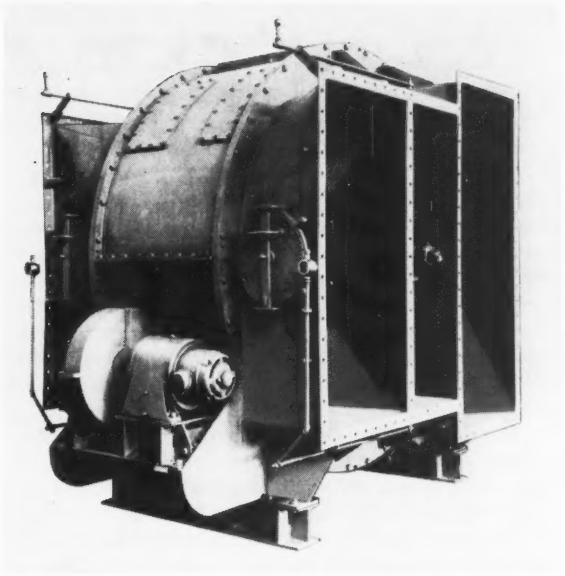

Fig. 3. Standard "H" Type Rotary Regenerative Air Preheater

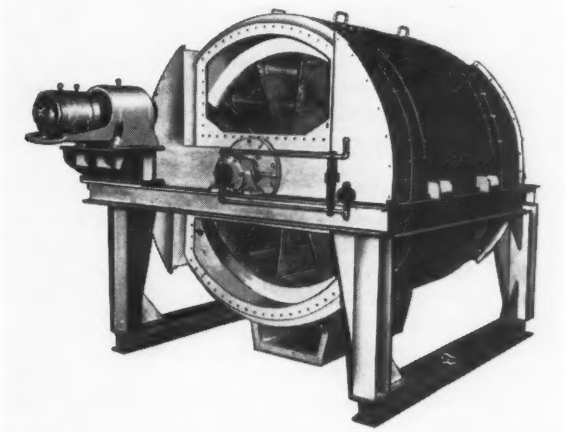

Fig. 4. High Temperature "H" Type Rotary Regenerative Air Preheater

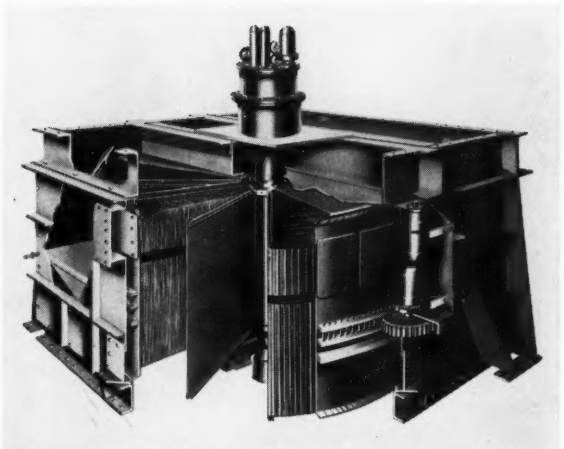

Fig. 2. Marine "V" Type Rotary Regenerative Air Preheater

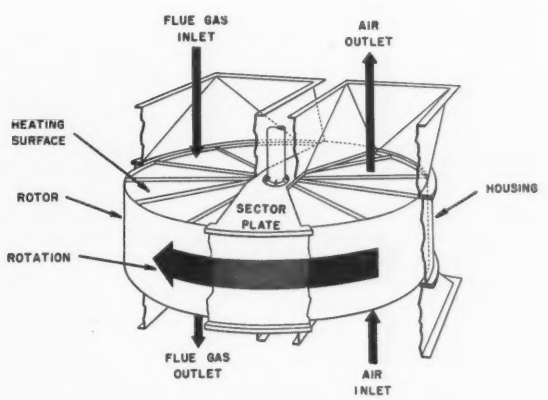

Fig. 5. Details of Rotary Regenerative Heat Exchange

types of machines. However, some unusual operating conditions exist and it is the combination of these which makes the lubrication problem a little more than ordinary. This paper will endeavor to discuss these unusual conditions and illustrate a few design features of unique interest.

CONDITIONS OF APPLICATION

A condition of the utmost importance is *continuous operation.* This equipment often must function without interruption for periods of several months, to a year or more. Once the unit is in operation, there is little opportunity for servicing unless an emergency arises to force the complete stoppage of the equipment. For this condition alone, lubrication assumes a role of major responsibility.

Another condition to recognize is the range of *operating temperatures.* Table 1 gives temperatures for several typical applications. Note, the range between the air inlet temperature and the gas inlet temperature for each application. These temperatures are by no means static. Variations often occur upward or downward because of changes in the duty placed on the equipment. Note, also, the elevated temperatures to which this equipment can be subjected. Units have been applied to entering gas temperatures as high as 1500 F, with abnormal variations upward in temperature sufficient to cause the destruction of certain metal components. These two factors—variations in operating temperatures and elevated temperatures—make necessary a number of special features to assure continuous and adequate lubrication.

The *speed of rotation* is an interesting condition. Rotation is normally in the range of 1 to 4 RPM. On occasions, and for short durations, the speed may be reduced to about 1/15 RPM to permit washing the heating surface. Variations are rare but must be taken into account when designing support bearings and lubrication facilities. Slow rotation does not permit the usual movement of oil and the effects of stagnation must also be anticipated.

The equipment is compactly designed to save space and weight. *Radiated and conducted heat* creates another condition of some significance. The trunnion, housing and lubricant must be adequately protected to offset this condition.

Ambient air conditions with respect to temperature and entrained matter in the area of the equipment are another factor to be considered. The equipment may be indoors or outdoors, with ambient temperatures varying from sub-zero to 150 F and above. Through the day, temperatures may vary 50 to 75 F depending on the arrangement and location of the equipment. The amount of dust and dirt entrained in the surrounding atmosphere varies widely depending on the location and condition of the equipment.

One cannot overlook *bearing load* as a condition of major consideration. The load is represented by the rotor. Typical bearing loads are given in Table 1. Loads are reasonably constant and rarely vary to any critical degree. An exception may result from structural deformations as temperatures are applied. Sufficient distortion could cause abnormal conditions effecting bearing alignment. Design considerations, based on experience, anticipate these conditions and provide necessary compensating features in the form of self-aligning bearings or plates.

Aside from mechanical design considerations, there is the very important matter of *selecting the lubricant.* While there is need for some precision in determining lubrication requirements, it is in the best interest of economy to select popular grades, readily available at local centers of distribution and in general use in the plant for other equipment. By so doing, the lubricants for the entire plant are standardized for the convenience of inventory and, more importantly, there is less chance of applying the incorrect lubricant.

The major areas of lubrication can be identified as:

1. Rotor support and guide bearing
2. Rotor drive speed reducer, electric motor and auxiliary air motor or steam turbine
3. Cleaning device speed reducer, electric motor and rotary joint

Rotor Support Bearings

The designs of the vertical and horizontal units are sufficiently different to warrant individual treatment. Figs. 6 and 7 illustrate two vertical assemblies. In vertical units, the entire rotor weight is carried on a single thrust bearing of the conventional ball or roller type or a Kingsbury shoe type bearing, depending on the size and arrangement of the equipment.

TABLE 1.—Temperatures and Bearing Loads for Typical Rotary Regenerative Air Preheater Applications

Boiler Evaporation—1000 lbs/hr	2,923	1,550	525	40	REVERB. FUR.	OIL STILL	FUSION FUR.	INCINERATOR
Type of unit	V	H	V	H	V	H	H	H
Temperatures:								
Gas entering (°F)	700	780	740	550	1100	1225	1400	1500
Gas leaving (°F)	311	268	340	312	232	410	420	500
Air entering (°F)	160	88	110	80	60	60	60	250
Air leaving (°F)	659	652	560	314	860	936	1000	1150
Location—indoor/outdoor	in	out	out	in	out	out	in	in
Weight of unit—1000 lbs.	721.8	393.5	74.3	10.1	112.0	40.7	16.8	133.5
Weight of rotor—1000 lbs.	510.9	313.8	52.6	5.9	77.0	23.6	9.5	31.8
Heating surface—1000 sq ft.	283.7	148.0	23.6	1.5	37.0	15.3	5.3	20.8

V = vertical flow
H = horizontal flow

The equivalent horizontal bearing assembles are illustrated in Figs. 8 and 9. In horizontal units, the load is divided between two bearing assemblies at opposite ends of the rotor. This type unit is equipped only with ball or roller type bearings since the shoe type does not readily lend itself to this type of unit arrangement.

Bearings are selected with respect to type and capacity to support loads from approximately 4,500 to 540,000 lbs for the largest unit currently available. The Kingsbury thrust bearing, limited to vertical units, is applied when the load exceeds approximately 110,000 lbs.

Most bearings are of the spherical roller, self-aligning type. For designs which do not include this type of bearing, the self-aligning feature is obtained with a spherical washer or plate assembly above the bearing. This is an exclusive feature of vertical units and is illustrated in Fig. 12.

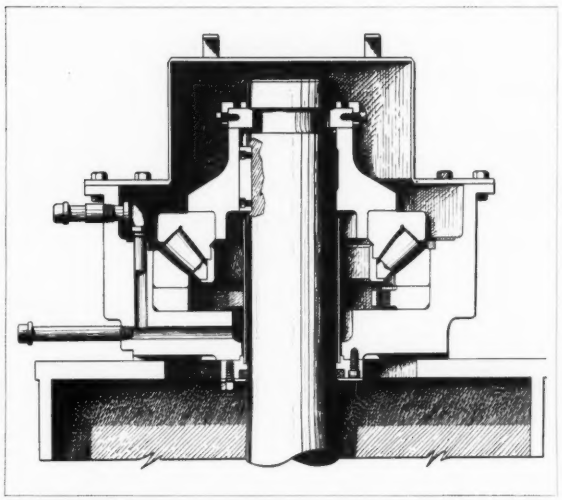

Fig. 6. Top Support Bearing Assembly for a Vertical Type Air Preheater

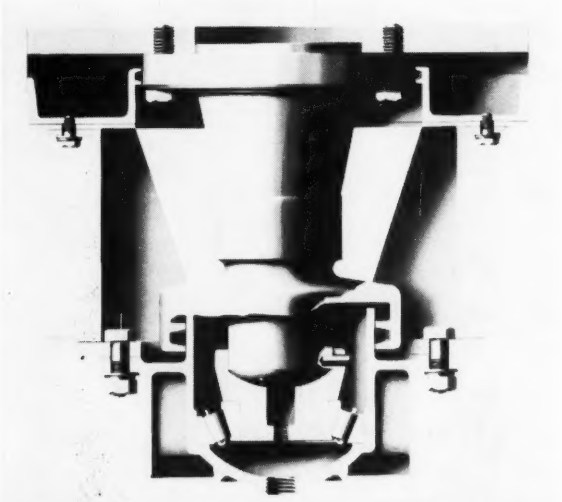

Fig. 7. Bottom Support Bearing Assembly with Spherical Seat for Vertical Type Air Preheater

Guide Bearings

Figs. 10 and 11 illustrate two guide bearing assemblies required in vertical type units to steady the rotor and to resist the pendulum effect which results from an unbalance in the flow of gases and air in opposite directions through the air preheater. Fig. 12 illustrates the relative position of the support bearing and guide bearing in a vertical type unit.

The guide bearing, usually a double row spherical roller design, is never heavily loaded. However, it is often subjected to the hottest conditions of operation, being situated directly under the air preheater in bottled-up heat and high ambient air conditions. When the hot gases enter and the hot air leaves the preheater at the bottom of a vertical unit, the radiation and conduction effects are most intense in the guide bearing area.

This may be a good point to draw attention to the fact that freedom for axial expansion of the rotor is provided by arranging one bearing assembly to float or slide in its housing. For example, in vertical units the load resting on the thrust bearing permits expansion downward with the guide bearing free to slide in its housing. For horizontal units, the same freedom is provided by securing one bearing in its housing and allowing the opposite bearing to float or move in its housing, or arranging the housing to float on rockers or expansion plates. This is one of the novel design considerations of the equipment.

Cooling

With the bearing assemblies situated between hot ducts and exposed to generous amounts of radiated and conducted heat, some means of cooling the lubricant

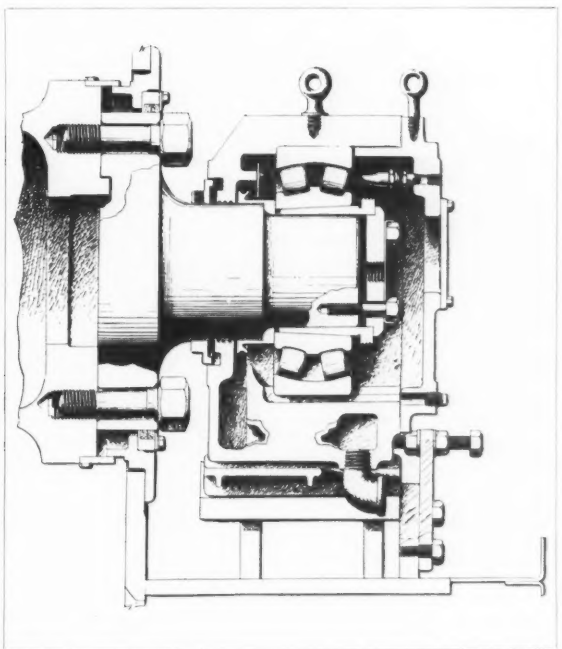

Fig. 8. Solid Trunnion Design for Horizontal Type Air Preheater with External Oil Circulating System

and bearing is necessary. Water and/or air cooling is usually employed. The shortage of water, in an increasing number of instances, is forcing the use of air cooling. The quantity of water required is not considered high for the worst conditions encountered but in certain situations even a small amount of water is a serious burden. Normal requirements call for about 3 gpm with the maximum not exceeding 23 gpm.

In some designs, the housing is jacketed for the circulation of water. This is usually the case for small and lightly loaded equipment. For example, the guide bearing housing in Fig. 10 is jacketed. The immersion of a cooling coil or box has been adopted on occasions.

For certain horizontal units, the trunnion is cooled by introducing water through a rotary joint as shown in Fig. 9. This feature is usually required for high temperature units where, due to the short distance between the bearing assembly and the hot interior of the air preheater, the amount of conducted and radiated heat may be sufficient over extended periods of operation to effect the bearing and lubricant.

The air cooling system usually consists of a standard industrial type circulating fan about 24 in. diameter and driven by a ¼ HP 1750 RPM electric motor. The lubricant passes through a radiator cooled by a blast of air. Its efficiency depends on a cool source of clean air and a distribution that will assure the removal of heat in sufficient quantity.

Lubrication

Lubrication for the support and guide bearing is important enough to require some unique considerations from the mechanical design point of view as well as in the selection of suitable lubricants. The position of the bearing housing between hot ducts and influenced by radiation and conducted heat creates an interesting problem for the lubrication engineer.

The ideal solution appears to be a self-contained lubricating system. This can be external to the bearing housing or an integral part of the assembly. In either case, cooling, filtering and circulating means are provided to protect the lubricant and the moving components. Figs. 13 and 14 illustrate two systems currently applied.

Support bearing assemblies illustrated in Figs. 6 to 10 inclusive indicate the bearing is immersed in an oil bath. In vertical designs, the bath is of generous dimensions permitting submersion of the bearing in a large quantity of oil varying from 2½ to 43 gallons depending on type and size of bearing assembly.

In horizontal bearing assemblies and guide bearing assemblies, the oil bath is less generous but still considered adequate for the lubrication required. The quantity for horizontal designs varies between 1½ and 11 gallons. For guide bearings about 2 gallons is the maximum required.

The external system in Fig. 13 has certain advantages. Major components can be placed in a cool location with easy access for control and maintenance. Unfortunately, it is almost impossible to eliminate leaking oil and spillage, particularly from around the shaft of the circulating oil pump. The unsightly condition gives advantage to the integral system with submerged pump.

The integral system practically eliminates the leakage nuisance. All components are built into the bearing housing cover resulting in a compact assembly. However, it has the disadvantage of being situated between hot ducts and not readily accessible for examination and maintenance.

The choice of lubricants depends on the expected maximum operating oil temperature of the support bearing which in turn, depends on the type of lubricating and cooling system, and on conditions surrounding the support bearing. Experience indicates temperatures tabulated in Table 2 may be expected for the several different types of air preheaters available.

The lubricant may be either an EP (extreme pressure) type with lead naphthenate additive, or a straight mineral base oil. If the oil operating temperature exceeds 160 F, it is considered wise to investigate the effect of such temperature on an EP lubricant.

Operating temperatures exceeding 180 F are viewed with some caution. Additional cooling or extra insulation on adjacent structure and duct work, or a combination of both, may be necessary.

The lubricant should be selected initially with SSU values noted in Table 3 to assure a minimum of 1,000 SSU at the expected operating oil temperatures in Table 2. Actual operating oil temperatures may dictate a change in lubricating oil to conform with the recommended minimum viscosity.

TABLE 2.—Expected Oil Bath Temperatures (°F) in Different Types of Air Preheaters with Consideration for Methods of Lubrication and Cooling.

	COOLED	
	WATER	AIR
Bath Lubrication		
Inverted-vertical unit without rotor support beam	200	160
Inverted-vertical unit with rotor support beam	140	120
Vertical unit without rotor support beam	140	120
Vertical unit with rotor support beam	120	100
Horizontal unit at hot end	200	160
Horizontal unit at cold end	140	120
Oil Circulating System—with or without cooler		
Inverted-vertical unit without rotor support beam	140	
Horizontal unit at hot side	140	
All other units and locations	120	

TABLE 3.—Recommended SSU Values for Anticipated Oil Operating Temperatures

SSU @ 100 F	SSU @ 210 F	OIL OPERATING TEMP. (°F)
1900–2200	124	100–120
3500–4000	178	120–140
8000–8600	275	140–160
21500–22500	460	160–180

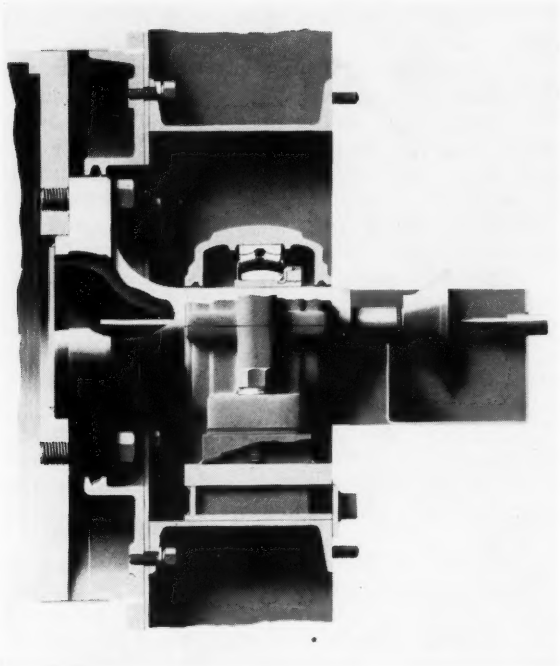

Fig. 9. Hollow Water Cooled Trunnion Design for High Temperature Horizontal Type Air Preheater

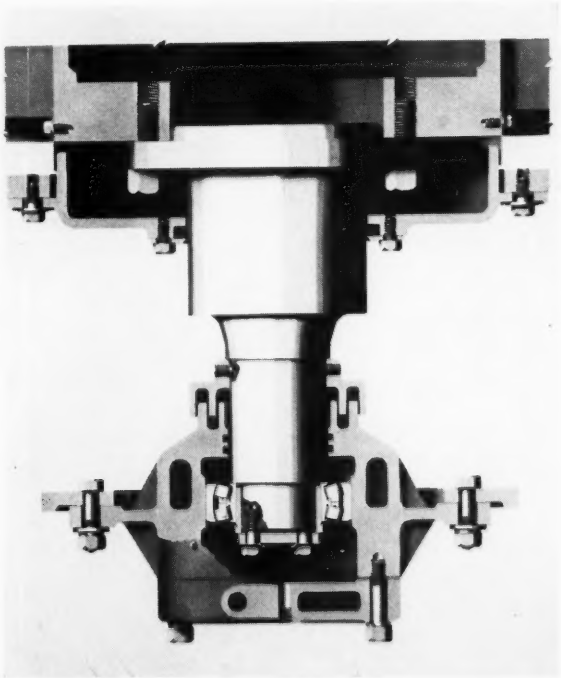

Fig. 10. Bottom Guide Bearing Design with Water Cooled Housing and Cover for Vertical Units

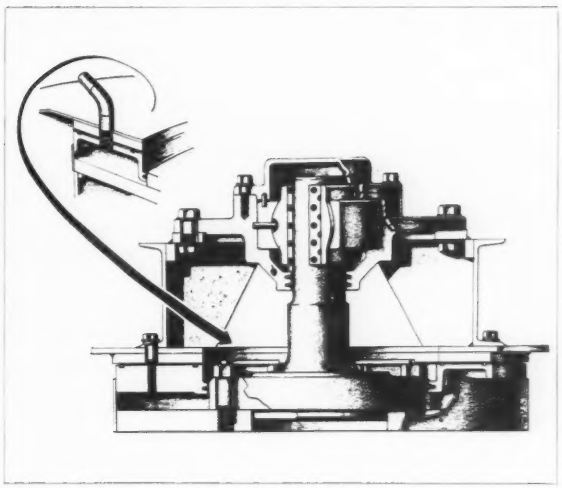

Fig. 11. Top Guide Bearing Design with Grease Impregnated Sleeve Bearing for Vertical Units

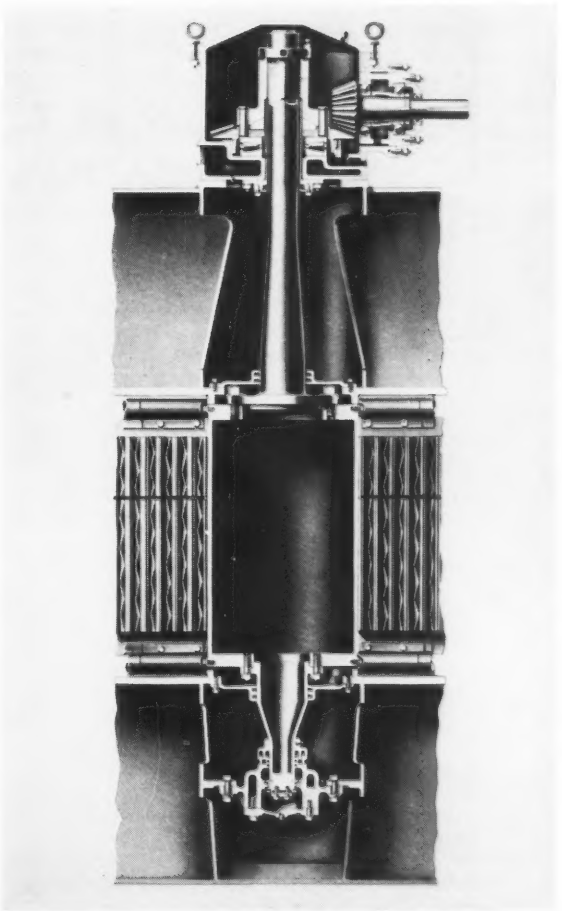

Fig. 12. Cross Section Through Vertical Unit Showing Rotor Support and Guide Bearing Assemblies and Rotor Details

The lubricant selected for the vertical support bearing is considered suitable for the guide bearing.

To assure the maximum in lubrication efficiency, it is the practice to drain the system after six months operation, flush out with light oil and refill with fresh oil. After this initial treatment, the practice is repeated on an annual basis or during convenient outages of about that period.

For units equipped with an oil circulating pump, it is recommended the viscosity of the lubricant not exceed that anticipated for an oil temperature of 140 F. This will avoid overloading the pump when the temperature is low, as for example, during the starting-up period. However, to guard against such overload, a thermostat with a sensing device in the lubricating oil bath is generally provided with oil circulating systems to limit operation of the system to oil of reasonable viscosity and to preclude operation of the system under abnormal conditions, such as with extremely heavy oils during the starting-up period.

Rotor Drive assembly

The rotor drive assembly assumes a number of arrangements, depending on the type of air preheater, position of duct openings, and size of unit. Figs. 15 and 16 illustrate assemblies for vertical and horizontal units respectively.

The electric drive motor is usually of standard design from ½ to 15 HP. The standard electrical characteristics result in speeds which have to be reduced appreciably and for which purpose a speed reducer is provided. The speed reducer may be of the spur, herringbone, helical or planetary gear design.

The position of the assembly places it at the periphery of the rotor, remote from hot areas adjacent to the preheater. Natural cooling with ambient air is usually adequate.

The quantity of lubricating oil is generous and depends on the type and capacity of the speed reducer. The small units require about 1½ gallons and the larger sizes take 12 to 14 gallons. The lubricant is a high grade straight mineral oil in consideration of the speed of the several gear systems—sometimes triple reduction—required for the desired output speed and the ambient conditions in the area of the installation.

The speed reducer manufacturer specifies the requirements over a range of conditions. Fortunately, they do not vary too much in the requirements because of adherence to AGMA (American Gear Manufacturers Association) standards. A typical recommendation is given in Table 4.

For such parts as may require grease lubrication, the recommendation is to apply a good grade of ball and roller bearing grease. Grease must be mechanically and chemically stable and suitable for this type of service.

In any case, the lubrication requirements for the size of equipment involved represents no great quantity of oil and no need to deviate from the widely accepted and available grades of lubricants. The air motor, for instance, is equipped with an air line lubricator, introducing oil into the air stream continuously, lubricating all the internal moving components except for the gear box which is grease lubricated.

For normal operating temperatures at the level of 60-120 F, oil mist lubrication is adequate with a high quality lubricant of approximately 150 to 200 SSU at 100 F (SAE 10). Selections to 1000 SSU at 100 F are possible but rare in the experience of this equipment. The lubricant must be free of graphites, soap and filler compounding.

TABLE 4.—Typical Lubrication Specification

TABLE (A)—AMBIENT TEMPERATURES

ROOM TEMP.	SAE NO.	VISCOSITY AT 100 F
15- 50 F	30	450–600
51-110 F	40	750–1000
Over 110 F	50	1200–1500

TABLE (B)—LUBRICANT VISCOSITY SPECIFICATIONS

APPROXIMATE SAE NUMBER	AGMA LUBRICANT NUMBER	VISCOSITY IN SUV AT 100 F	210 F
10	1	180–240	—
20	2	280–360	—
30	3	490–700	—
40	4	700–1000	—
50	5	—	80–105
60	6	—	105–125

A list showing typical oils meeting the above specifications can be found in Appendix A to the AGMA Standard 250.02 on Specifications for Lubrication of Industrial Enclosed Gearing.

THE LUBRICATING OIL USED IN ENCLOSED "AP" GEAR DRIVES MUST MEET THE FOLLOWING SPECIFICATIONS:

1. High quality, well refined petroleum oil.
2. Noncorrosive to gears, ball, roller or sleeve bearings and free from grit and abrasives.
3. Good defoaming properties, and good resistance to oxidation.
4. Straight mineral type.
5. Viscosities in conformance with Table (B) to right.

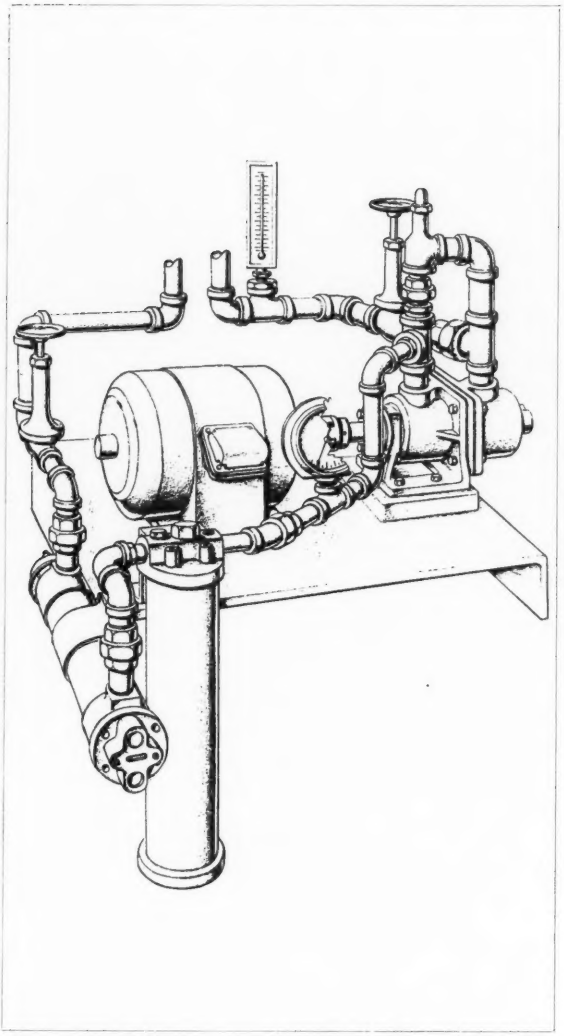

Fig. 13. External Type of Lubricating Oil, Circulating, Filtering and Cooling System

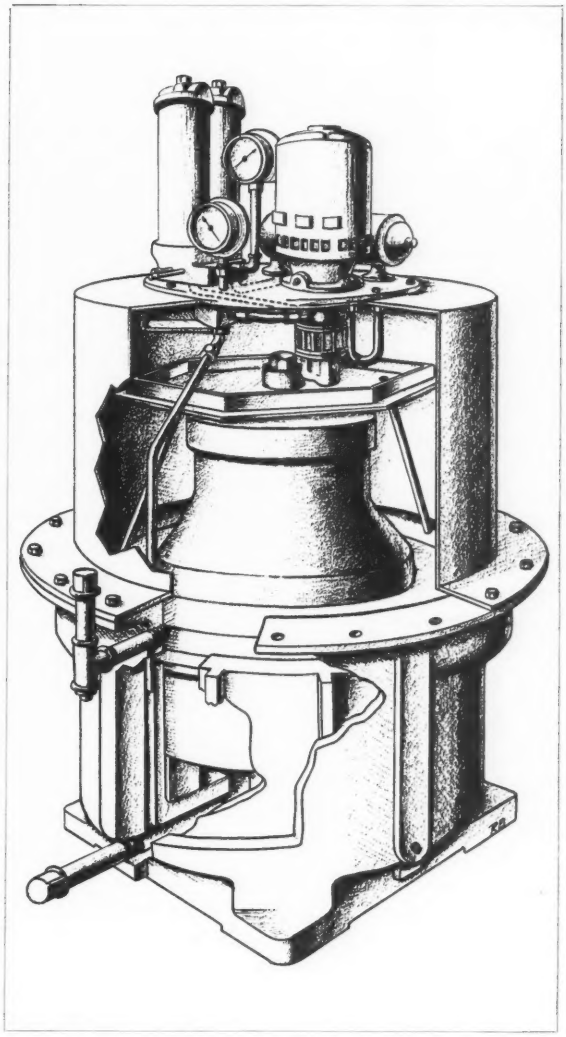

Fig. 14. Integral Type of Lubricating Oil Circulating, Filtering and Cooling System

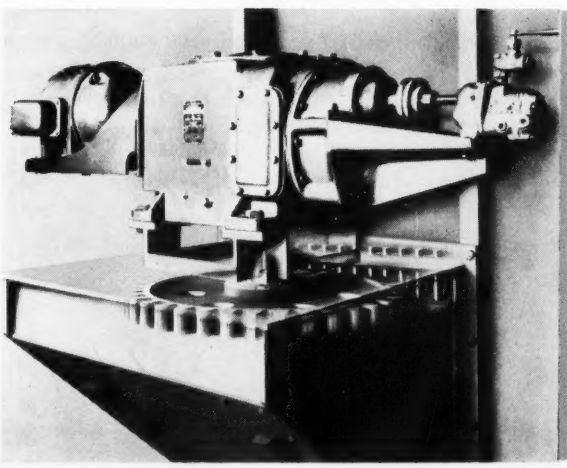

Fig. 15. Rotor Drive Assembly for Vertical Units

Cleaning Device

The cleaning device with its several operating components is considered one of the most important features of the rotary regenerative air preheater. The oscillating nozzle pipe, actuated by a mechanism external to the air preheater distributes steam, air or water over the heating surface matrix to remove any accumulation of soot and dirt. The principal design is illustrated in Fig. 17. It is a compact assembly and requires only a modest amount of lubrication. Due to the slow rate of oscillation of the nozzle pipe, about 0.5 feet per minute—a large reduction in speed is provided in the form of a small speed reducer. Since the duty is light, a ¼ HP electric motor is sufficient when coupled to a speed reducer of standard design using worm and worm-wheel members to achieve the 10,000 to 1 reduction desired.

Lubrication recommended is a high grade noncorrosive type of EP oil. Although the equipment is situated remote from the hot areas of the air preheater it is subject to varied ambient temperature conditions depending on whether the air preheater is part of an indoor or outdoor plant. For this reason, the following ranges are observed:

Ambient Temp. (°F)	*Lubrication*
0-35	SAE 90
40-110	SAE 140

The large worm and worm wheel following the speed reducer of the cleaning device moves at such a low rate as to make lubrication no problem. A heavy gear oil or semi-fluid grease is more than adequate.

The final item of lubrication is the rotary joint in the steam, air or water line to the cleaning device. This is an important ball bearing item, exposed to cold or hot water when washing the heating surface matrix and to steam at temperatures as high as 900 F. A most significant point is the joint never makes a full revolution but rather oscillates through an arc not exceeding 90 degrees. The exposure to hot steam or water and the lack of rotation to distribute the lubricant produces what might at first appear as a difficult lubrication problem. Actually, the amount of lubricant required is exceedingly small and occasional inspection assures satisfactory operation.

For the water handling joint, a small amount of grease is introduced through an alemite fitting to protect the bearing from corrosion. For the steam joint no lubrication is required since the bearing races and balls are treated with a dry film lubricant adequate for the life of the equipment.

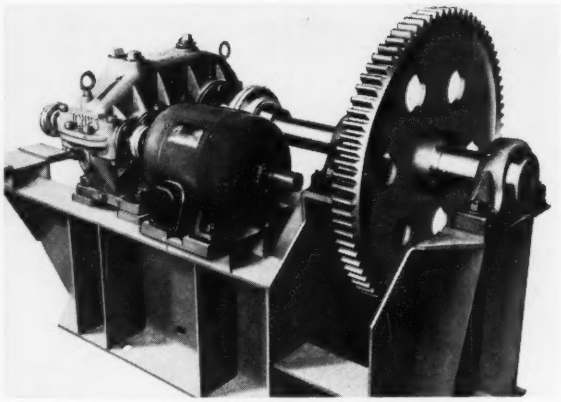

Fig. 16. Rotor Drive Assembly for Horizontal Units

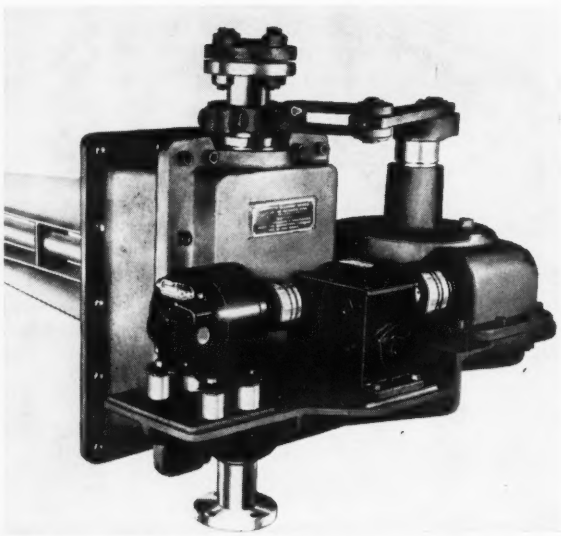

Fig. 17. Single Nozzle Power Operated Oscillating Type Cleaning Device

CONCLUSION

In spite of the unusual operating conditions with respect to slow rotation, high temperatures and related radiation, conduction, expansion and distortion, and heavy bearing loads, lubrication for the rotary regenerative air preheater is not a complex design problem or difficult maintenance function. Adequate facilities have been provided to meet all design requirements without resorting to extreme measures and innovations. Maintenance has been reduced to a routine function with periodic inspection and renewal of the lubricant to suit the conditions of operation.

HYDRAULIC FILTER REQUIREMENTS

of Industrial Servo Systems

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This paper discusses filtration requirements of industrial electro-hydraulic servo control systems. It points out the critical areas in wet and dry coil sevro valves, and discusses the type of failures due to contamination and their consequences for the systems involved.

Filtration media characteristics are discussed in terms of the requirements of a reliable servo control system. Methods of measuring these characteristics are reviewed. Degree of filtration, nominal efficiency, absolute cut-off, bubble point, dirt capacity, media migration, built-in-dirt, abrasion, channeling, collapse, and element construction are discussed.

Types of depth and surface filter media and their characteristics as related to servo systems, are discussed.

Features and limitations of filter housings are described, and the modular mounting of filter elements in servo packages with differential pressure indicators is discussed and illustrated.

The report concludes with recommendations for filter elements for the protection of servo control systems.

Presented at the 15th ASLE Annual Meeting in Cincinnati, Ohio, April, 1960

INTRODUCTION

The need for filtration in both industrial and military hydraulic systems has long been recognized. The main controversy has always centered on the amount of contaminant which must be removed, or how fine a filter must be to assure economical and reliable operation.

The heart of the newest generation of machine tools is the electro-hydraulic servo control systems, and with their advent filtration requirements have had to be re-evaluated. The degree of filtration which was generally satisfactory for older type of industrial hydraulic systems has proven inadequate for systems containing electro-hydraulic servo valves.

All servo failures are not necessarily caused by contaminant. They can also be caused by hydraulic, mechanical or electrical malfunctions; but contaminant is often the suspected or proven cause of the following types of system or component failures:

(a) Hard-over movement or locked position of the servo-controlled actuator
(b) Positional error or oscillations of the actuator
(c) Actuator velocity unbalance for the two directions of travel
(d) Transient jump of actuator
(e) Increase of servo-valve null leakage with age
(f) Scored actuator piston rods
(g) Scored valve seats
(h) Failure of pumps and scored pump parts
(i) Fatigue failures of hydraulic lines and components
(j) Jammed pumps and valves

A hard over movement in a control cylinder can be destructive in a hydraulically controlled machine tool or in many other types of servo-controlled industrial hydraulic systems. A positional error or transient jump in an automatic machine tool cutting head can be costly in both rejected parts and potential machine damage.

THE SERVO VALVE

For the purpose of this paper, contamination problems associated with the two-stage electro-hydraulic servo valve will be considered.

Servo Valve Types

Fig. 1 shows examples of two types of two-stage servo valves: dry-motor and wet-motor. Not shown is the stagnant motor type. The first stage of each valve is similar in that it contains an electric torque motor which drives a flapper which, in turn, regulates the flow through a single or double orifice. The differences in the first stages are in the environment of the magnetic gap of the electric torque motor. In the dry-motor type, the magnetic gap is isolated from the hydraulic oil and contains air. In the wet-motor configuration, oil is present in the magnetic gap and there is a small flow of oil through the gap. In the stagnant-motor configuration, oil is present in the gap but there is no flow. The second stage is basically the same in all three types of valves; it consists of a spool with appropriate lands in a ported sleeve. The flow through the ports is regulated by the position of the spool in the sleeve, which, in turn, is regulated by the position of the flapper (1).

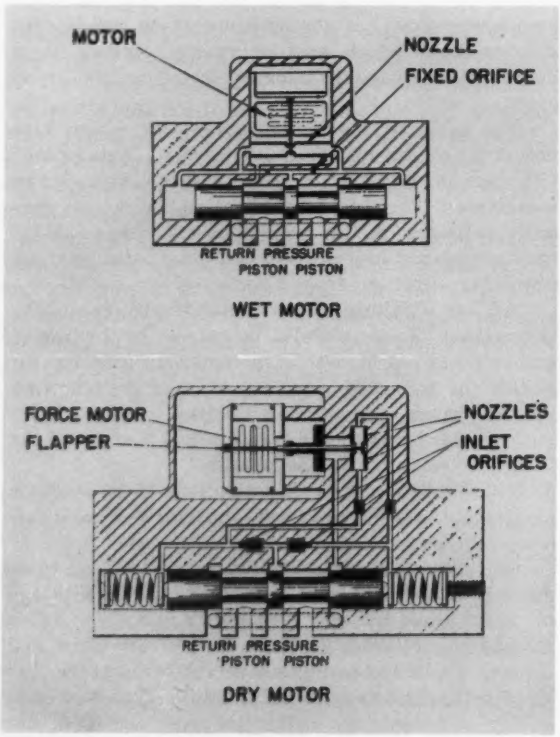
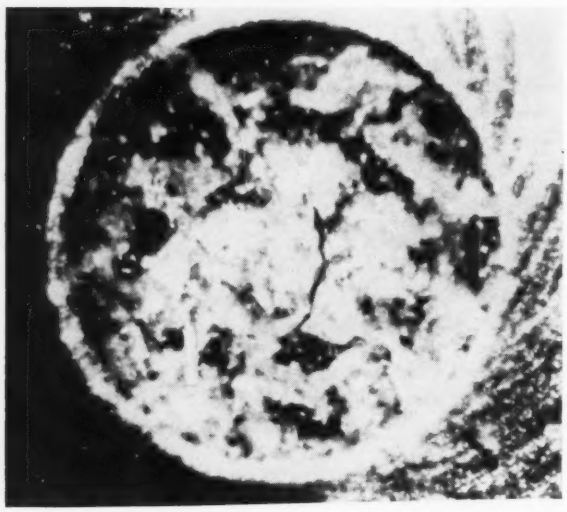


Fig. 1. Wet- and Dry-Motor Servo Valves

PLUGGED NOZZLE EXTERNAL
ENLARGED 10 TIMES



PLUGGED NOZZLE EXTERNAL
ENLARGED 45 TIMES

Fig. 2. Servo Valve Nozzle Blocked by Fibers from Hydraulic Oil

Critical Areas

It is not the purpose here to discuss the over-all merits of one type of servo valve as compared to another; however, some observations are pertinent. From the standpoint of sensitivity to contaminant, the following areas of an electro-hydraulic servo valve are critical:

(a) Nozzle—Normally, nozzle diameters range from 100 to 500 microns. Any particle larger than the nozzle diameter will partially or completely block the nozzle causing a hard-over signal in the actuator, a valve unbalance which can cause positional error in the control system, or a velocity unbalance in the actuator velocity for the two directions of travel. The nozzle diameter is large enough to prevent blocking in all normal cases where adequate filtration is available. However, Fig. 2 shows how, in a specific instance, fibers present in hydraulic oil created a "log jam" over the nozzle which, in turn, retained small particles which ultimately blocked the nozzle orifice

(b) Flapper—The clearance between the flapper and the passage is considered the most critical area in the first stage, and measures approximately 25 to 50 microns. A flapper held immobile by contaminant locks the actuating cylinder. If a particle is large enough to move the flapper to its extreme position, a hard-over signal is produced in the actuator. A particle passing the flapper can briefly prevent normal flapper movement or can create a small unwanted flapper movement. The effect of either is a transient movement of the actuator. In a servo-controlled machine tool such a transient erects a monument to itself on the work piece.

(c) Spool and sleeve—The diametral clearance between the spool and sleeves is in the range of 1 to 6 microns. When the valve is operating the flows are high, and combined motion and flow normally flushes the contaminant through the spool and sleeve. The forces are generally high enough to shear normal size particles caught in the annular space, but high concentrations of abrasive contaminant can erode the spool and sleeve, ultimately changing their characteristics. However, if the spool is held in a stationary position with pressure applied, the spool sleeve radial clearance acts as a very fine (absolute ½ to 6 micron) filter for the oil leaking through the annular clearance. The resulting build-up of dirt, called silting, creates a pressure unbalance, thereby causing an increase (sometimes quite marked) in breakout-friction, called "sticktion". These pressure unbalances can cause serious oscillations in the servo system. This is indicated by a drift followed by a correction when the amount of drift is sufficient to create an error signal strong enough to overcome the sticktion. This cycle repeats itself. Since sticktion occurs only in the stationary position, one way to avoid it is to create a small oscillatory movement of the spool; this is called "dither"

(d) Magnetic gap in wet-motor servos—The magnetic gap acts as a filter for magnetic contaminant in the oil. As these particles build up they change the characteristics of the electric torque motor, and ultimately they jam the flapper, causing the actuator to lock.

FILTRATION REQUIREMENTS

What characteristics must a filter possess to assure reliable servo valve operation and how are these characteristics to be evaluated?

Degree of Filtration

An ideal filter should remove all insoluble contaminants from the hydraulic fluid. As the size of the particles becomes finer, the number of particles increases markedly and the cost of removing them also increases. Absolute cleanliness, like an absolute vacuum, can be approached but never attained.

Since it is not practicable to remove all of the insoluble contaminant from a hydraulic fluid, removal limits for particle number, size and type must be defined. Also, for wet-motor servo valves, special attention must be paid to the removal of magnetic particles. To assure that the limits set are maintained, test methods have been devised to measure the degree of filtration.

Filter ratings are normally given in microns. A micron is one millionth of a meter and is equal to 0.000,039 inches (39.37 millionths of an inch). To comprehend the approximate magnitude of a micron, the average diameter of human hair is 75 microns; the red corpuscle of human blood has a diameter of approximately 7.5 microns; and the finer synthetic organic fibers are 15 to 30 microns in diameter.

Nominal Ratings

Nominal ratings have been generally used in industry and are also defined in Military Specifications MIL-F-5504A and B. A 10-micron filter is defined by MIL-F-5504A as being capable of removing 98% by weight, of all particles exceeding 10 microns of the selected contaminant, U.S. Army Standardized Air Cleaner Fine Test Dust, at a certain high concentration. Mil-F-5504B defines a 10-micron filter as being capable of removing 95% by weight of 10-to-20 micron glass beads at a certain high concentration (2). Two loopholes exist which make the use of filters which meet only these requirements unsatisfactory for the protection of servo valves. The first is that the maximum size of the 2% fine test dust or 5% glass beads which is allowed to pass through the filter element is not limited. Tests by many laboratories have found that nominal 10-micron filters which meet the requirements of MIL-F-5504A and B readily pass particles in the range of 100 to 200 microns (3). The second loophole is that the contaminant concentrations used in the test simulate the high concentrations of a total pump failure and are not at all indicative of performance under normal system conditions. With normal concentrations, the efficiency at 10 microns is much less than that determined by the specification test.

Until the introduction of all-stainless-steel sintered woven wire mesh filter elements to the aircraft industry in 1955 all degree-of-filtration ratings were nominal (4). The uniformity of the hole size of the fine sintered woven wire mesh has made possible the adoption of absolute ratings.

Absolute Rating

Absolute filtration ratings are defined by proposed Military Specification MIL-F-8815 and proposed MIL-F-25682 (USAF) (5, 6). The rating is determined by measuring the diameter of the largest glass bead to pass through the filter. It is equivalent to the point at which a filter is 100% efficient. Although the standard tests are run using glass beads, other tests have shown that the maximum glass bead passed is approximately the same size as the maximum dimension or normal system contaminant passed. The absolute filter rating is not affected by changes in contaminant type or concentration: thus it is a very useful parameter.

A very simple inspection method, the bubble point test, is used to measure the maximum particle passed by a filter and thus determine the absolute rating of the filter. Fig. 3 shows how this test is conducted. The filter element to be tested is immersed in a specified fluid (normally denatured alcohol, Federal Specification O-E-760b, Grade 4, but hydraulic oil can be used with less accurate results) and air pressure is applied to the inside of the immersed filter element as the element is rotated slowly. The first continuous stream of bubbles locates the largest hole and the pressure at which it is produced is a measure of the size of this largest hole. Fig. 4 shows the correlation between the bubble point pressure of a filter element and the maximum particle passed through the filter element. This particular curve is applicable to sintered stainless steel wire mesh filter elements. Slightly different curves are obtained for other filter media (in most cases differing by not more than 10%).

Magnetic Efficiency

In addition to the foregoing, wet-motor servo valves require the use of magnetic filters. To protect a servo valve adequately a magnetic filter should be able to remove and retain magnetic particles in the range of one micron and larger. Analysis of contaminant removed from magnetic filters after field use show large open-type structures; however, when a demagnetizer is passed over the microscope slide the large open particles disintegrate into thousands of small (1 to 5 micron) particles. This shows that in a magnetic field very small particles can agglomerate to form large particles. The value of a magnetic filter is that it removes these small particles before they reach the servo valve.

Dirt Capacity

A filter must exhibit adequate on-stream life without servicing. The service interval is dependent upon the amount, size and type of contaminant, the flow rate, and the condition of the hydraulic oil. Since actual contamination varies from system to system, it is not possible to simulate exact service life by test. Comparative service intervals can be obtained by measuring the amount of fine test dust required to raise the

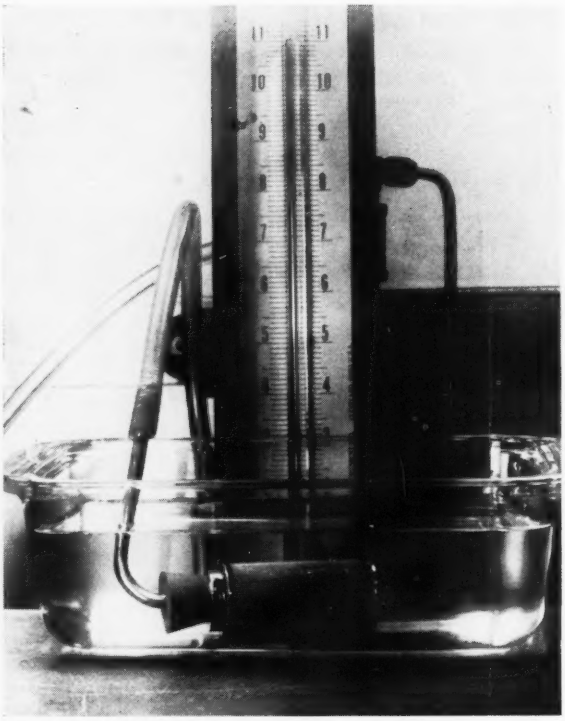


Fig. 3. Bubble Point Test and Test Apparatus

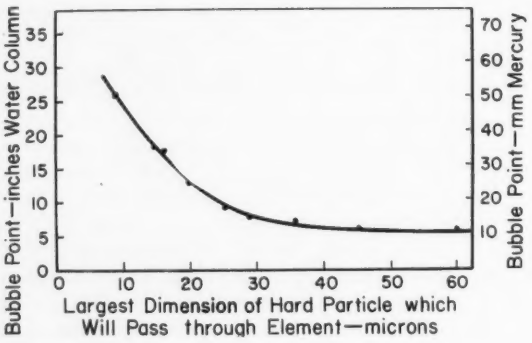


Fig. 4. Correlation between Maximum Particle Passed and Bubble Point for Sintered Stainless Steel Wire Mesh, Wet with Denatured alcohol, Federal Specification O-E-760b, Grade 4

differential pressure of various filters to 90 psi or to some other arbitrary value. A typical dirt capacity curve is shown in Fig. 5. The form of this curve, in which the increase in pressure drop is initially very slow, is characteristic. A filter which is 50 percent clogged may have a pressure drop of only a few tenths of a psi more than its clean pressure drop. Such a small differential cannot be detected with the ordinarily available measuring equipment. Thus, a high pressure drop indicates a dirty filter element, but a low pressure drop does not necessarily indicate a clean filter element.

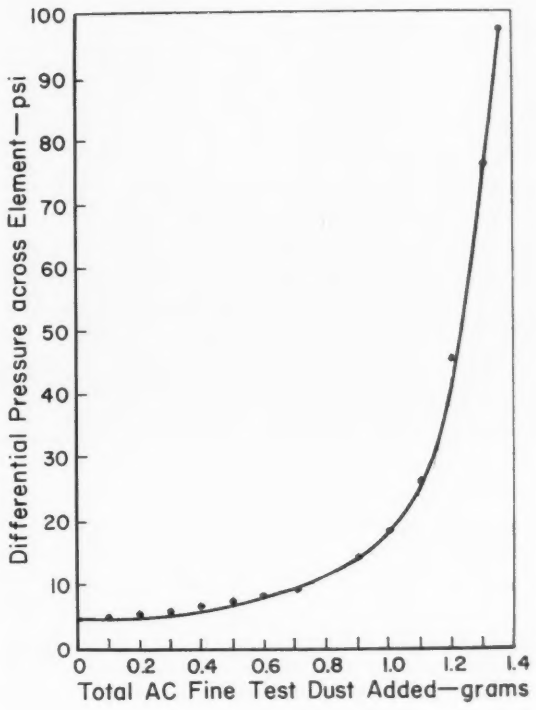


Fig. 5. Typical Dirt Capacity Curve for Hydraulic Filter Element

Migration

Another requirement of a servo-valve filter element is that it should not release contaminant downstream of the element. The processes of contaminant release are of four basic types:

The first type is called media migration and occurs when part of the filter medium breaks away and adds itself to the system contaminant. This can happen with or without a major breakdown of the filter media. Media migration is to be expected with all types of filter media made up of fibers. To protect a servo valve against such migration, fiber media such as paper, wool, felt, cellulose, glass fiber, etc. should either be eliminated or else followed with a nonmigrating-type filter of sufficient fineness to remove the fiber migration.

This problem applies to a lesser extent to filter elements constructed of sintered powder. Perfectly bonded sintered powder should never migrate, but to date no nondestructive quality-control test has been devised which will guarantee perfect bonding. Thus, to protect a servo valve from potential migration, sintered powder filters should either be replaced or followed with a nonmigrating type filter of sufficient fineness to stop all migration.

Lack of media migration is inherent in the design of a woven wire mesh filter. It is made up of long strands of wire compressed and locked together in a tight weave. Wires can be removed only at the edges and from around imperfections. When the mesh is sintered even these possible sources of migration are eliminated since all wires are welded at all points of contact. Thus, sintered woven wire mesh can be considered completely free of media migration.

The second type of contaminant release from the downstream side of a filter element is called "built-in-dirt migration." All types of filter elements, made from all media, suffer from the disadvantage of having dirt released in use which was trapped in the filter element during manufacture. The amount of "built-in" dirt can be reduced greatly, but cannot be eliminated completely, by means of improved production techniques and special cleaning during and following manufacture. A filter element manufactured with no special care paid to cleanliness can release very large quantities of contaminant into the system. Proposed Aeronautical Recommended Practice 599 (7) has been developed to measure built-in-dirt.

The third type of contaminant release is called abrasion migration. Abrasion particles may be generated when a filter element is inserted in its housing or in a vibration environment between a poorly designed filter element and filter housing. Most of the generated contaminant is on the outside of the element and acts only to shorten the service life of the element. However, part of the contaminant generated between the element seal and the housing is on the downstream side of the filter element. Filters for servo valves in a vibration environment must be specially constructed to eliminate this problem.

The fourth and final type of release is called contaminant migration. It is applicable only to fiber-type media and other media that change porosity under varying conditions of flow and pressure drop. Under such conditions, and this includes almost all servo systems, contaminant stopped on the upstream side of the filter media slowly works its way through the media and is ultimately released on the downstream side. This problem is eliminated by replacing such media with a rigid medium such as sintered woven wire mesh.

Structural Strength

A servo valve filter assembly must be constructed to withstand all structural and operational loads to which it will be subjected over its entire temperature and life range. The filter housing must be capable of withstanding repeated pressure cycles without fatigue; it must also withstand mounting and wrenching loads.

The filter element must be designed to give proper support to the filter media and must have sufficient

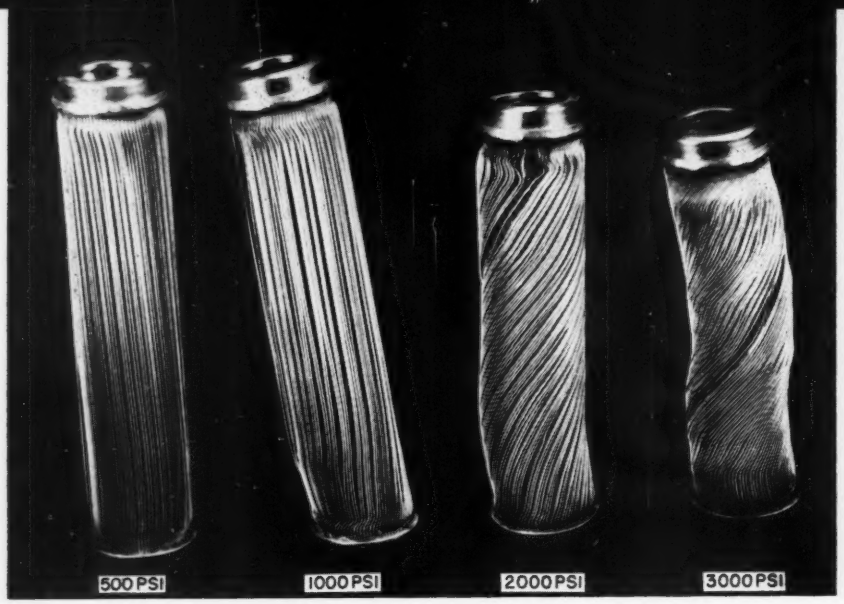


Fig. 6. Effect of Excess Differential Pressure on Filter Elements

column strength to withstand the differential pressure developed across the fully dirty filter element. Fig. 6 shows the effect of exceeding the design differential pressure (which in this case is 350 psi) on sintered woven wire mesh filter elements. The filter element must also maintain an adequate seal between the filter medium and the hardware throughout the useful differential pressure range of the filter element. Several designs of stacked-disc-type metal elements are on the market which allow the mesh to deflect from the outer and inner sealing rings as differential pressure increases, thereby creating an undesirable bypass.

Any filter medium for servo valve protection should have an unalterable uniform fixed hole size to prevent channelling and contaminant migration. The filter media must also be capable of withstanding the constant cycling differential pressure resulting from flow fluctuations common in servo systems without developing fatigue cracks.

Compatibility with Hydraulic Fluid

A servo-valve filter must not affect or be affected by the hydraulic fluid at either the high or low operating extremes of temperature. Some types of filter media slowly dissolve or disintegrate in certain hydraulic fluids. Dissolution of the resin binder of a certain type of paper element has been held responsible for some of the difficulties recently experienced in a new commercial aircraft. Other types of media have the ability to remove corrosion inhibitors, viscosity improvers, or other additives from hydraulic fluids.

TYPES AND CHARACTERISTICS OF FILTER MEDIA

Filter media can be broken down into two general categories for purposes of discussion: depth and surface. Paper, felt, cellulose, glass fiber, wood pulp, diatomaceous earth, and sintered powders are generally used as depth media; they depend upon a relatively long tortuous path to remove contaminants. Wire mesh and membrane are generally used as surface media where dirt is removed by a plane surface containing fairly uniform orifices. Thus, most if not all of the contaminant is on the surface of the medium. All media act to varying degrees as both surface and depth media.

Fiber Media

Fiber media such as paper, felt, cellulose, and glass, are available in varying fiber diameters, densities, and thickness. Nominal filter ratings are available from ½ to 100 microns. Absolute ratings are not normally provided due to the non-uniform nature of the media, but if measured, generally vary from 10 to 50 times the nominal ratings.

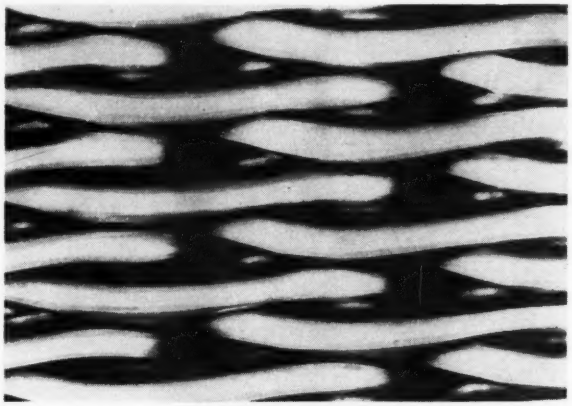
Sintered Powders

Sintered powders generally act as depth media. They can be made in a variety of materials, including most metals and ceramics and many plastics. Sintered stainless steel offers the advantages of inertness to hydraulic fluids and high strength. Filtration properties range from nominal 2 to 65 microns and are varied by changing the size of the powder. Absolute ratings can be controlled and vary from 13 to 100 microns. The major disadvantage of sintered powders is media migration. Proper manufacturing techniques can eliminate this problem, but any relaxation of quality control standards can permit areas of poor bonding which will result in potential or actual media migration.

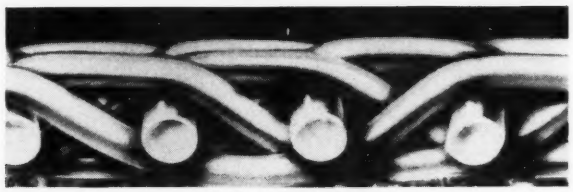
Woven Wire Mesh

Woven wire mesh is available in a number of weaves. Those most commonly used are square, such

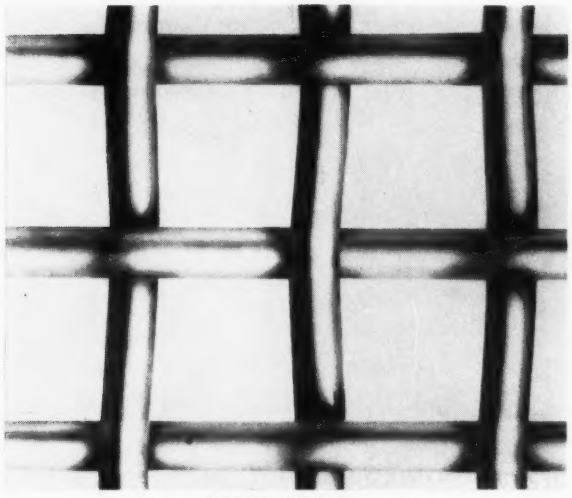
as window screen and finer, and dutch twill, as shown in Fig. 7. Wire meshes are very uniform. Filtration ratings vary from nominal 2 to 100-plus microns. Wire mesh possesses good strength, and since it is made of long wires, is free of media migration. Sintered wire mesh has the added advantage of rigidity and fixed hole size. It has low dirt capacity per unit area, but its thinness allows the use of high area configurations. For a given installed volume, wire mesh often has more dirt capacity than a depth medium. The absolute ratings can be easily controlled and vary from 12 to 200-plus microns.



DUTCH TWILL WEAVE
(TOP VIEW)



DUTCH TWILL WEAVE
(SECTIONAL VIEW)



SQUARE WEAVE

Fig. 7. Woven Wire Mesh Weaves

Membrane

Membrane is a plastic filter medium composed primarily of cellulose nitrate. Fig. 8 shows an artist's impression of the actual structure of membrane. Only absolute filter ratings are given for membrane; they vary from 5 millimicrons to 12 microns absolute. The medium is monolithic and is thus free from migration. It has very low dirt capacity and poor structural strength. It is particularly useful in laboratory analysis of oil contaminants, but is also finding use in high and low-pressure oil filtration for critical systems. The commonly available membranes dissolve in the phosphate ester-based hydraulic fluids and they are therefore limited in their industrial applications.

Combinations

To attain both the greatest dirt capacity and the most efficient removal of contaminant from hydraulic oil, combinations of surface and depth media are recommended. The depth medium provides dirt capacity and removal of fine particles; the surface medium used downstream of the depth medium provides for absolute control of maximum particle size and the retention of media and contaminant migration from the depth medium.

A recently developed combination medium using a fine grade of glass fiber followed by membrane, provides long life and single-pass absolute 3-micron filtration.

Another new combination is sintered stainless steel woven wire mesh with a sinter-bonded stainless steel powder facing. A photomicrograph is shown in Fig. 9. This absolute 15 micron all-stainless-steel combination is already providing a five-fold dirt capacity increase in existing hydraulic systems using the existing filter envelope and housing with the same absolute and same or better nominal filtration.

Magnetic Filters

The magnetic filter element of a typical airborne high efficiency magnetic filter is divided into two

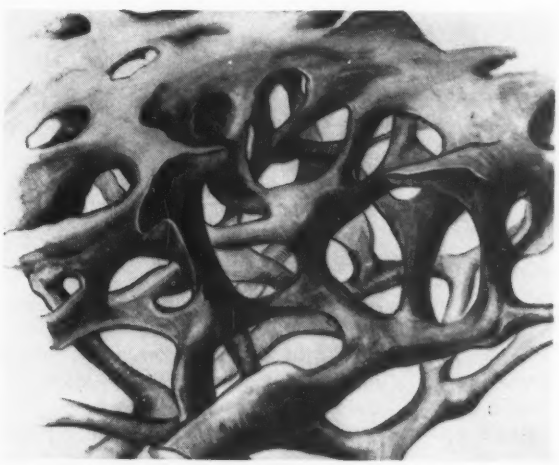


Fig. 8. Cellulose Nitrate Membrane

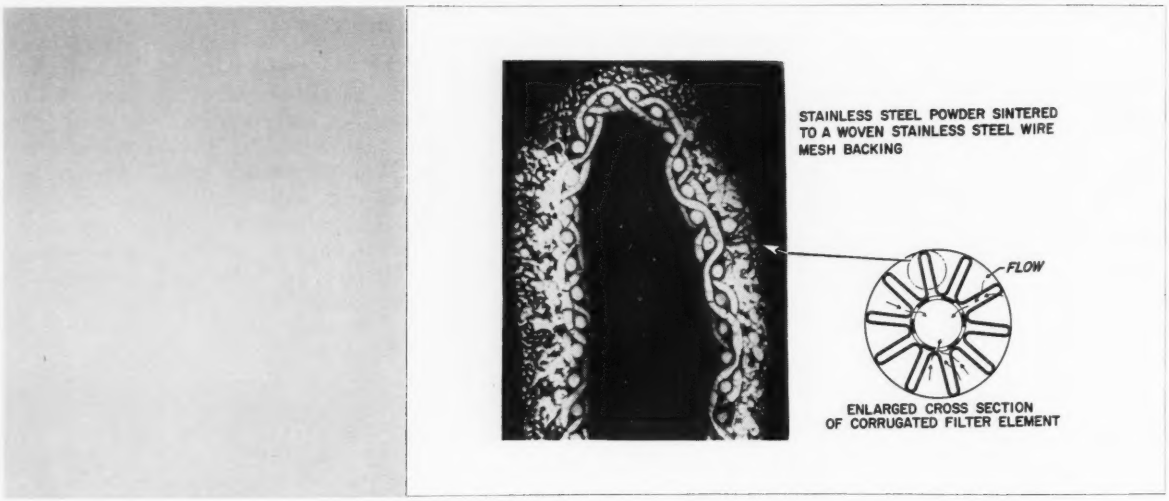

STAINLESS STEEL POWDER SINTERED TO A WOVEN STAINLESS STEEL WIRE MESH BACKING
FLOW
ENLARGED CROSS SECTION OF CORRUGATED FILTER ELEMENT

Fig. 9. Etched Sample of Sintered Woven Wire Mesh with Stainless Steel Powder Facing

stages. The first stage is a depth magnetic filter that is designed to pass all of the hydraulic fluid through an area of high magnetic flux. The small magnetic particles collect on the surfaces of the magnets and grow into large particles. The second stage is a sintered woven wire mesh surface filter. It collects normal system contaminants and the larger magnetic particles if and when they grow too large and break off from the magnetic stage.

The intensity of the magnetic field is greatly increased by filling the gaps between the large magnets with corrugated specially woven wire mesh. In this mesh, the wires parallel to the axis of the magnets are paramagnetic. This increases greatly the magnetic field in the gap since the wires become induced magnets. The wires perpendicular to the axis of the magnet are nonmagnetic, which prevents a magnetic short circuit.

THE FILTER HOUSING

The filter housing must be designed to withstand the structural loads of installation, use, and system environment.

Relief Valve

Maximum filtration reliability is obtained when there is no relief valve to bypass a filter element, and nonbypass housings are recommended for servo system applications. However, when dictated by other system requirements, relief valves can be incorporated into a small compact housing, as shown in Fig. 10.

Automatic Shut-off

An automatic shut-off is recommended for filter housings used in servo systems. This prevents air inclusion, and eliminates drainage of part or all of the system when a filter element is changed. The elimination of loss of fluid is of particular importance when using the more expensive fire resistant hydraulic fluids. Fig. 10 shows the automatic shut-off in both open and closed position.

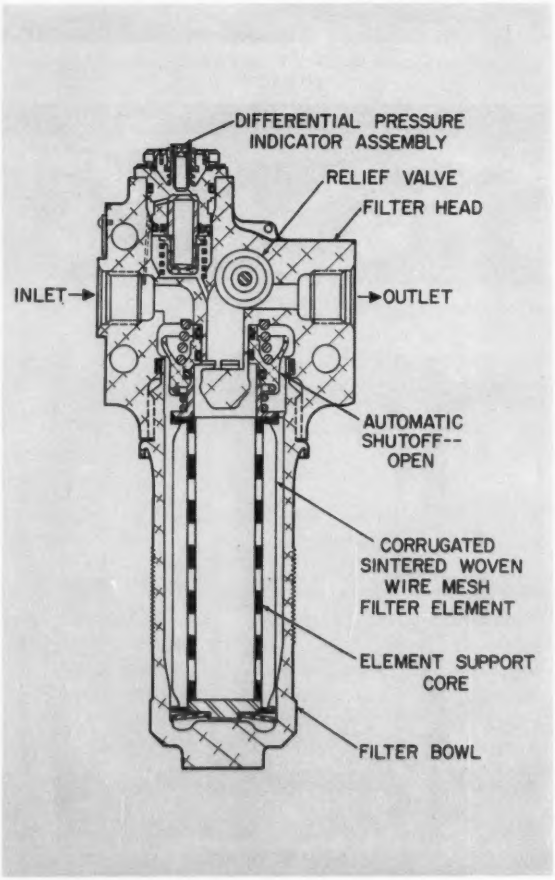


Fig. 10. Aircraft Hydraulic Filter Assembly

Differential Pressure Indicator

Fig. 10 shows a cross-section of a magnetic differential pressure indicator mounted on top of a filter housing. When differential pressure causes the lower magnet to move to a predetermined position, the upper spring extends the upper magnet and raises the red button. No external dynamic seals are required. Such a device can save countless maintenance hours and prevent unnecessary opening of the hydraulic system. The button can be modified to actuate a switch for those systems requiring an electrical indication.

Modular Installation

The new trend in military servo systems is the modular component system. Fig. 11 shows a modular assembly for a missile containing a filter with differential pressure indicator, servo valve, and actuator in one package. This package has the advantages of: reduced size and weight; increased reliability through the reduction of lines, fittings, and seals; easier maintenance; and close proximity of the filter to the servo valve.

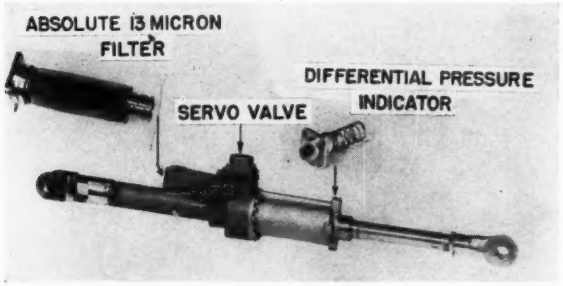


Fig. 11. Modular Assembly of Filter, Servo Valve and Actuator

RECOMMENDATIONS FOR FILTER MEDIA TO PROTECT A SERVO VALVE

For industrial applications the optimum filter medium is a glass fiber and membrane combination rated at 3 microns absolute. This type of filter is available in high-dirt-capacity high-pressure configurations, which allows it to be used immediately upstream of the servo valve. This close proximity to the servo valve provides the best possible protection. For systems using fluids which are not compatible with membranes a properly supported absolute 3-micron fiberglass medium should be used. In either case an absolute 15 micron sintered woven wire mesh back-up filter is recommended.

For optimum operation, a servo system should be assembled with precleaned components that are absolute 3-micron pressure-flushed immediately before installation. The assembly area should be dust controlled. Dummy servo valves should be installed initially and the system should be absolute 3-micron pressure-flushed (with turbulent flow where possible) until the contaminant content, as measured per ARP 598 (8), levels off. The servo valves should be installed and the flushing continued until the contaminant level again stabilizes. The system is then ready for use. This flushing should be repeated any time the system is opened. Any new or repaired components should be precleaned and flushed immediately prior to installation.

CONCLUSIONS

The emphasis of this paper has been on the need for careful evaluation, specification, and selection of filter media, elements, and housings in relation to the requirements of those critical areas of electro-hydraulic servo systems which are particularly sensitive to contamination.

In general, the cleaner the hydraulic oil, the more reliable the servo system. Therefore, for greatest reliability, a servo system should be assembled of precleaned components and should be flushed thoroughly immediately prior to use with precleaned oil. The system filtration should be rated at absolute 3 microns to assure removal of the majority of particles in the stiction size range, and the filter should be located as close to the servo valve as possible.

It is also essential that the system cleanliness level and detailed filter requirements be included as part of the original specifications for electrohydraulic servo control systems. In this manner, proper coordination of contamination control with performance requirements will be achieved.

While it is true that many manufacturers pride themselves on designing components and systems that will "eat dirt," few, if any, of the designers of such items will deny that any component or system will perform more reliably and last longer when contamination is controlled.

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Performance of

EP GEAR LUBRICANTS

*in the rear axles of an over-the-road truck fleet**

by W. C. Brandow,
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*Article Published in the April, 1961 issue of *Lubrication Engineering*

DISCUSSION

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In general we are prompted to agree that the data presented, tends to support the idea that the failures being encountered are more probably traceable to a combination of the axle size employed and the operating conditions prevailing in this fleet, rather than the type of lubricant being employed in the axles. In any event it does appear true that the performance of the three lubricants tested was approximately equal.

We note that the lubricants used in these tests were an SAE-90 viscosity grade. In this connection our overall experience indicates that we get better performance with an SAE-140 lubricant than we do with an SAE-90 lubricant. This is particularly true in an operation such as the one described and involving problems substantially identical to those encountered here. This naturally leads us to wonder whether or not an SAE-140 viscosity lubricant in any one or all three of the types tested, might have led to better performance.

We note that in several instances, where gear failure or deterioration was involved, it is indicated that the bearings appeared to be serviceable; and, in one case, it is indicated that the bearings are in good condition except for small indentations from foreign particles. In this connection, it is our experience that it is almost impossible for the bearings to be this healthy in the presence of the wear noted on the gearing. Even though the bearings might otherwise appear to be in good condition, we normally find that the bearing roll ends have been lapped to an appreciable extent.

Fig. A1 illustrates an advanced stage of this condition. The wear on the roll ends is accompanied by an equivalent amount of wear on the thrust shoulder of the bearing cone. This wear is usually more pronounced on the pinion bearings and naturally leads to looseness or endplay in the bearings. In the absence of information to the contrary, we are wondering whether or not the units in question had roll end wear and endplay in the pinion bearings prior to disassembly.

The report indicates that eight out of the fourteen pinions that are still in service have a "line of small cavities in the surface near the edge of the contact pattern at the root of the tooth". Although not specifically covered, it is implied that the gears are in good condition. Our experience is that this pitting develops at the actual pitch line and accordingly, we refer to it as pitch line pitting. The fact that these pits are definitely inside of the edge of the contact pattern indicates that they cannot be at the base line; and accordingly, we are prompted to rule out base line interference. Our experience is that this condition results from regular though intermittent metal-to-metal contact between the gear and the pinion.

Since the normal sliding action on the face of the pinion is away from the pitch line in both directions, the intermittent metal-to-metal contact causes a slight

depression to form at the pinion pitch line. As the condition progresses, small pits will begin to appear in this depression. Continued operation will cause these pits to enlarge and eventually small pieces will flake off in between several pits. This will be followed by metal flaking off of the areas adjacent to the pitch line and a general growth of the spalled area.

The action on the mating gear is quite different. The normal sliding on the gear tooth face is toward the pitch line from both directions. Accordingly, the intermittent metal-to-metal contact will cause a high spot to develop at the pitch line. The presence of the high spot on the gear aids and abets the action that is going on in the pinion because the high spot tends to increase the instantaneous pressures.

Fig. A2 illustrates the conditions under discussion at a moderately advanced stage. Looking at the second pinion tooth from the top, we can observe a flaked-out area which is just above the lower boundary of the tooth contact. Looking at the uppermost pinion tooth, we can see what appears to be a depressed area extending to the right in approximately the same region as the flaked-out area on the second tooth. Looking down at the gear teeth we can observe the raised area a slight distance below the tips of the gear teeth.

Although we have raised some questions which were not covered by the report, they do not alter our concurrence with the conclusion that the performance of the three lubricants appear approximately equal.

Fig. A2. Wear of Axle Gears

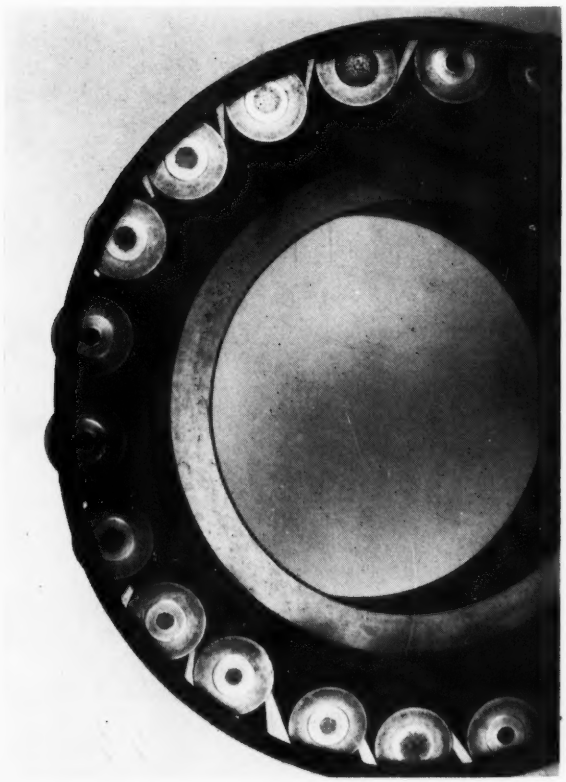

Fig. A1. Wear of Roller Ends on Axle Bearing

DISCUSSION

by R. K. Smith

The Elco Lubricant Corporation
Cleveland, Ohio

The Lubrizol progress report on the fleet test of axle lubricants is of great interest to all of us. The unexpected difficulties which arise on a program such as this increase greatly the chance of inconclusive results, especially when the important anchor of uniform oil changes is conveniently replaced with a variable interval which depends on the condition of certain drain samples.

It is unfortunate but typical of road tests of this sort that load variations do exist as great as the 29,500 to 51,300 pound variation shown in the paper. We feel that it is asking too much of pure chance to prevent the placing of these very light loads on a given axle at a critical point in its life. Furthermore, the terrain over which this fleet operates is so flat in contrast to the grades negotiated in the West that lubricants tested here will not necessarily provide the factor of safety required for heavy duty service.

Nevertheless, we do agree based on the data on Table 2 that at this point no difference in favor of any one lubricant exists. However, there are 14 axles still on test and the inspection data shown in Table 3 tell an interesting story.

We have taken the liberty of reproducing this table with a slight modification as shown in Table B1.

TABLE B1
DISTRIBUTION OF PINION FATIGUE

LUBRICANT TYPE	NO. OF AXLES IN SERVICE	NO. OF PINIONS WITH FATIGUE AT BASELINE	NO. OF PINIONS SHOWING NO FATIGUE
MIL-L-2105 (SCL)	4	2	2
MIL-L-2105 (SCP)	6	3	3
MIL-L-002105A (SCPZ)	4	3	1

The surface fatigue mentioned here is attributable to a lack of torque carrying capacity in the lubricant and is a progressive type of failure which compounds itself once it has begun.

From the slide it is evident that fatigue has begun in 3 out of 4 axles on the MIL-L-002105A lube and in 5 out of 10 on the MIL-L-2105 lube—a significant difference in favor of MIL-L-2105.

Much is made in the paper of the distress on the differential parting face with no explanation of how this could possibly affect the surface fatigue on the gears. Furthermore, axle manufacturers (3 to be exact) state unequivocally that this type of distress has nothing to do with surface fatigue of the gears and is common to all makes of single reduction axles.

In commenting on the laboratory axle test data presented, we feel that the reason for the existence of this type of test should be stated again. High torque, low speed tests fill the axle designer's need for data confirming the strength of his tooth design. This same test also gives him an idea of the factor of safety he can expect with the various types of lubricants. Like the engine oil oxidation tests these high torque tests are here to stay.

Returning to the laboratory test data shown in Table 5, let us consider the average test duration for each of the test lubricants. Ample justification on a statistical basis exists for stating that the O result on the MIL-L-2105 lubricant is not in the same family as the other three results. Therefore, the average test duration for the MIL-L-2105 is 55, 100 wheel cycles while that for the MIL-L-002105A is 38,200—an indication of things to come in the fleet test described in the paper.

In order to provide the customer (both commercial and military) with suitable lubricants, recognition must be given to tests conducted in axles larger than the L-37 axle according to procedures which demonstrate a factor of safety acceptable to the axle builder. When this condition is met the problem of gaining acceptance for new lubricants will be eliminated.

AUTHOR'S CLOSURE

As Mr. Johnson suggests, investigation into the performance of an SAE 140 grade oil versus an SAE 90 would be very interesting. However, the equipment manufacturer referred to in the paper recommends SAE 90 for the service conditions involved and the program was limited to this grade.

It is true that roller bearing end wear was evident both visually and as end play in the pinion shaft. The reference in the paper to serviceable bearings indicates that there was no gross distress in these bearings, such as fatigue, cracked rollers, etc. In view of the number of broken axle shafts and abrasive particles, it was not practical to make precise measurements to determine bearing wear in an investigation comparing oil performance.

The explanation of the mechanism of pitch line pitting helps clarify this point and indicates extensive investigation on the part of this discussor.

Mr. Smith's statistical and arithmetic treatment of data is unsound in both the case of the laboratory endurance test results and in the occurrence of fatigue in pinions. The limited data on both these items does not justify statistical handling of the data to determine the significance of differences in lubricant performance. Mr. Smith asserts without supporting evidence that "The surface fatigue mentioned here is attributable to a lack of torque carrying capacity in the lubricant . . . " where there are logical grounds for deducing that the fatigue may be induced by axle deflections under load.

The manufacturer of these axles has suggested that distress on the differential parting surface indicates deflection, and, since this component locates the ring gear, a question arises as to whether or not the gears remain in correct contact. As suggested in the paper, a change in the contact pattern probably develops high localized stress resulting in fatigue.

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Lubrication Abstracts

Edited by W. E. Campbell

Selected literature compiled by members of the Technical Committees and Industry Councils of the Society.

"Recent Development In High Temperature Bearings And Lubricants". P. C. Hanlon (Mechanical Branch, Aircraft Laboratory, Wright Air Development Division, Wright-Patterson Air Force Base, Ohio) SAE Paper no. 1768 for meeting Apr. 5-8, 1960, 44 p.

The result of recent contract research on rolling contact bearings in three categories are reviewed. These are: (1) High Temperature Airframe Bearings, (2) High Temperature Accessory Bearings, and (3) Medium and High Temperature Engine Bearings. In (1) a program for testing grease-lubricated, self-aligning, oscillating roller bearings was executed. The objective is to develop ultimately a bearing type which will be capable of operation at altitudes up to 250,000 feet and will be resistant to the effects of nuclear radiation. Also included was development of airframe oscillating ball bearings in the −65 to 1200°F range. The object in (2) is to develop a ball bearing design, a suitable lubricant, and a self-contained lubrication system for use in aircraft electrical accessories for operation in the 500-700°F range. One aspect consisted in the determination of the relationships of grease-fill to bearing life, temperature to grease life, and the determination of facts connected with lubricant failure. Future development of suitable lubricants calls for high temperature testing and then low temperature testing of such compounds as ethers, ferrocenes, and related groups. Another aspect was concerned with testing ceramic compounds. Some materials such as titanium carbide cermet were run in the "four ball" wear tester. In (3) studies were made on large angular-contact ball bearings at moderate speeds, heavy loads, and high temperatures, as well as on small angular-contact ball bearings operating at light loads, high speeds, and high temperatures. No significant advances in data connected with medium and high temperature engine bearing lubrication resulted from these tests. It was shown that for temperatures above 1000°F organic lubricants are ruled out. No single outstanding material for bearings operating in the 1000-1200°F range was found, but two materials were found to operate satisfactorily over limited periods at the 1200°F level using powdered lubricants applied by inert-gas jet. A general finding of these programs is that there is a strong possibility that because the development of bearing materials has far outstripped that of lubricants some types of bearings designed for service in the −65°F to 1200°F range and higher, will have to operate without lubrication. *(Abtractor: William Rosenberg)*

G. Gobbi and E. Melgara, "Quality Rating for Lubricant Greases", La Rivista RIV (Italy), v. 8, Jan. 1960, pp. 8-22 (In Italian), E.T.D. n. 3769.

Four test apparatus used in the RIV laboratory are described. Each simulates one of the mechanical effects of ball bearings on grease deterioration. (1) The fibre disintegrator consists of a cylinder inside which rotates a cage having rods and curved fins, the latter compel the grease to drop continually for disintegration by the rods. 200g. of grease are beaten for 10 min. at 1,900 r.p.m. After each of four such beatings the ASTM penetration is determined. (2) Penetration is determined after a 10 hr test with the three-ball tester, 5 hr at 300 r.p.m. and 5 hr at 600 r.p.m. (Cf. Abstract in Lubrication Engineering, v.16, n.3, 1960, p.122). (3) Vibration resistance is evaluated in an apparatus consisting of two small pots in one of which is placed 100g. of a reference grease and in the other 100g. of the test grease. The pots are vibrated at an amplitude of 8mm. at 2,000 vibrations per min. for one hr. ASTM penetration is measured. (4) The extrusion stability test apparatus consists of a steel cylinder inside which slides a piston; a 1 mm. circular space exists between the piston and cylinder. When the piston moves, the grease is extruded through this space. ASTM penetration is measured after 1,000, 2,000, 3,000 and 4,000 extrusions. A high-quality lithium grease showed little deterioration after each of these tests; a low-grade sodium grease showed marked increases in penetration values during each of the tests. *(Abstractor: Edward H. Loeser)*

D. C. Lemmon (General Electric Company, Lynn, Massachusetts) and E. R. Booser, "Bearing Oil-Ring Performance." *Jour. Basic Eng.* v. 82, Series D, no. 2, 1960, p. 327.

This experimental investigation establishes quantitative expressions for ring speed and oil delivery as a function of several operating and design parameters. Since ring speed has a very significant effect on oil delivery rate, particular attention is devoted to the determination of ring speed as a function of journal speed. The other significant parameters investigated include ring size, ring material, ring configuration, immersion depth, oil viscosity, scrapers, and ring material. It is found that oil ring guides and scrapers are the most important design factors studied with respect to ring oil delivery rate. The paper is particularly useful to bearing designers since it enables a determination of the ring speed to be made under various operating conditions. Knowing the ring speed, it is then possible from the expressions given in the paper to determine oil delivery rates as a function of design parameters and operating conditions. *(Abstractor: H. Aparian)*

N. Kendall, "Standardization of Lubricating Oil Engine Test Methods by the Institute of Petroleum," *Institute Petroleum Journal*, v. 45, no. 430, October 1959, pp. 299-310.

This article is a report of the Engine Tests of Lubricants Panel describing three inexpensive test engines, two diesel, and one gasoline powered. The features of these engines as well as the test procedures for each are described. The Gardner 1.L2 and the Petter AV.1 engines are intended to be used in place of the Caterpillar L1 engine. The Petter W1 gasoline engine is intended to replace the Chevrolet engine for Oxidation Tests. Test data are shown for all three engines as well as data comparing the Caterpillar L-1 test to the Petter AV.1 AT.4 tests. The data are not complete for comparison of the Chevrolet L4 engine tests with the Petter W1 engine tests. *(Abstractor: George C. Lawrason)*

JOHN BUTLER TYTUS *dreamed that steel could be rolled in a continuous strip. In 1904, fresh from Yale University, he went to work as a laborer in the rolling mill to learn the craft, rose to mill superintendent and worked out the techniques and blueprints of his continuous rolling mill during World War I. It took him until December, 1923, to build the first mill and another six months of trial operation to produce satisfactory sheets. Today, billions of dollars worth of smooth-finished, deep drawing sheets roll at ever-increasing speeds from the continuous mills of America.*

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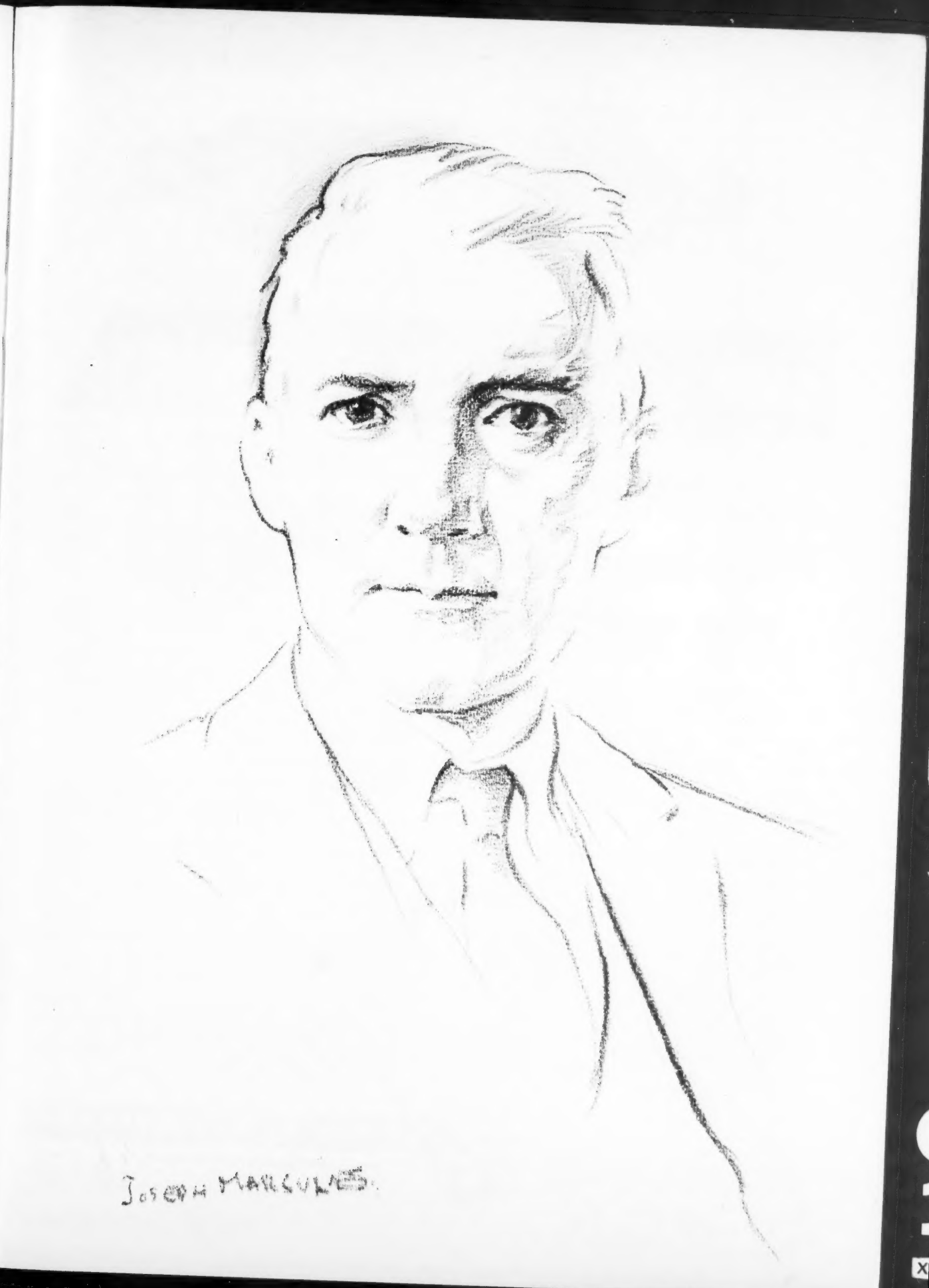
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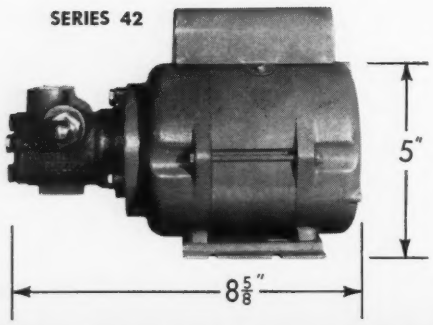

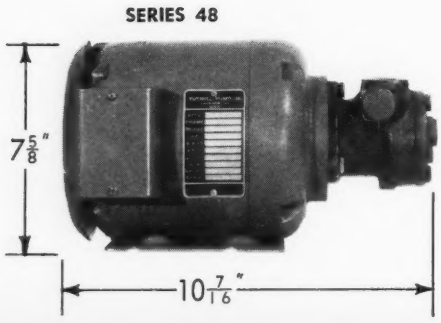

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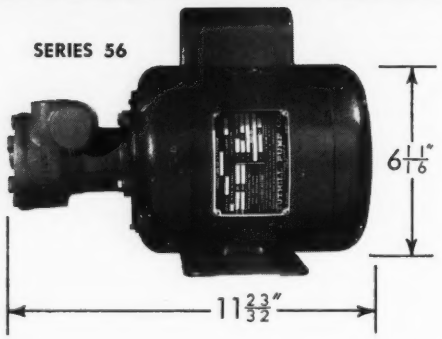

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Patent Abstracts

Compiled by Ann Burchick
Aluminum Co. of America

Wire Drawing Lubricant Containing a Diamide, Hydrated Lime, and a Normal Calcium Soap, Patent No. 2,956,017 (W. M. Franks, assignor to Nopco Chemical Co.) A wire drawing lubricant consisting essentially of a dry mixture of from about 5% to about 15% by weight of a diamide having a melting point above about 130 C., said diamide being the reaction product of ethylene diamine and a saturated aliphatic monocarboxylic acid having a carbon chain length of from about 12 to about 18 carbon atoms, and from about 85% by weight to about 95% by weight of a high calcium content composition comprising hydrated lime and a normal calcium soap of a saturated aliphatic monocarboxylic acid having a carbon chain length of from about 12 to about 18 carbon atoms, the hydrated lime comprising from about 35% to about 65% of the total weight of said high calcium content composition.

Marine Diesel Lubricant, Patent No. 2,944,-022 (C. J. Ulzheimer and J. C. Zimmer, assignors to Socony Mobil Oil Co., Inc.) An emulsion lubricant for marine diesels which consists of, by weight of the lubricant, between about 10 percent and about 50 percent water, between about 80 percent and about 45 percent mineral lubricating oil, having a Saybolt Universal viscosity of between about 45 seconds and about 100 seconds measured at 210 F., between about 1 percent and about 5 percent alkali-metal hydroxide, the amount thereof being sufficient to effect a total base number in the lubricant of between about 10 and about 100, between about 1 percent and about 10 percent calcium petroleum sulfonate, between about 3 percent and about 7 percent of a mixture of polyhydric alcohol esters of high molecular weight acids obtained by controlled oxidation of petroleum wax, and between about 0.5 percent and about one percent saturated fatty acid having between 16 and 20 carbon atoms per molecule.

Lubricant Compositions Containing Metal Salts of Aromatic Hydroxy Carboxylic Acids as Antioxidants, Patent No. 2,951,808 (J. H. Norton and W. C. Pattenden, assignors to Esso Research & Engineering Co.) A lubricating composition comprising lubricating oil and an oxidation inhibiting amount of a di-alkali metal salt of a hydroxy benzoic acid.

Process for Preparing a Lubricant and Composition Resulting Therefrom, Patent No. 2,955,122 (A. C. Whitaker, assignor to Gulf Research & Development Co.) A process for producing a lubricant composition from Oxo bottoms remaining after alcohol production wherein the olefin employed in the hydroformylation reaction stage had from 6 to 16 carbon atoms which comprises subjecting said Oxo bottoms to oxidation for a time sufficient to oxidize about 40 to about 60 percent of said Oxo bottoms and obtain an oxidation mixture comprising acids and alcohols, and thereafter reacting said acids and said alcohols at a temperature of about 60 C. to the boiling point of said mixture while the same are in said mixture.

Wax Composition Comprising a Mixture of Specific Paraffin Waxes and 1-2% of a Microcrystalline Wax, Patent No. 2,943,991 (J. D. Tench and I. N. Duling, assignors to Sun Oil Co.) A wax composition consisting essentially of from 45% to 54% of a paraffin wax having a melting point of from 131 F. to 136 F., a penetration at 77 F. of from 10 to 16, and a viscosity at 210 F. of from 35 to 41 SUS, and from 45% to 54% of a paraffin wax having a melting point of from 148 F. to 156 F., a penetration at 77F. of from 9 to 15, and a viscosity at 210 F. of from 40 to 46 SUS, and from 1% to 2% of a microcrystalline wax having a melting point of from 190 F. to 196 F., a penetration at 77 F. of from 3 to 8 and a viscosity at 210 F. of from 80 to 90 SUS.

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PERSONALS

(*Continued from page* 205)

Acheson Colloids Company has recently announced company plans to further expand the Applications Development Group within this firm. To supplement this program the services of **Mr. Roger H. Hendrick** have been acquired as an Applications Development Engineer. Mr. Henrick's business background covers 12 years as a Mechanical and Lubrication Engineer in various phases of metalworking industry. Mr. Hendrick will work with Acheson in a product development and research capacity.

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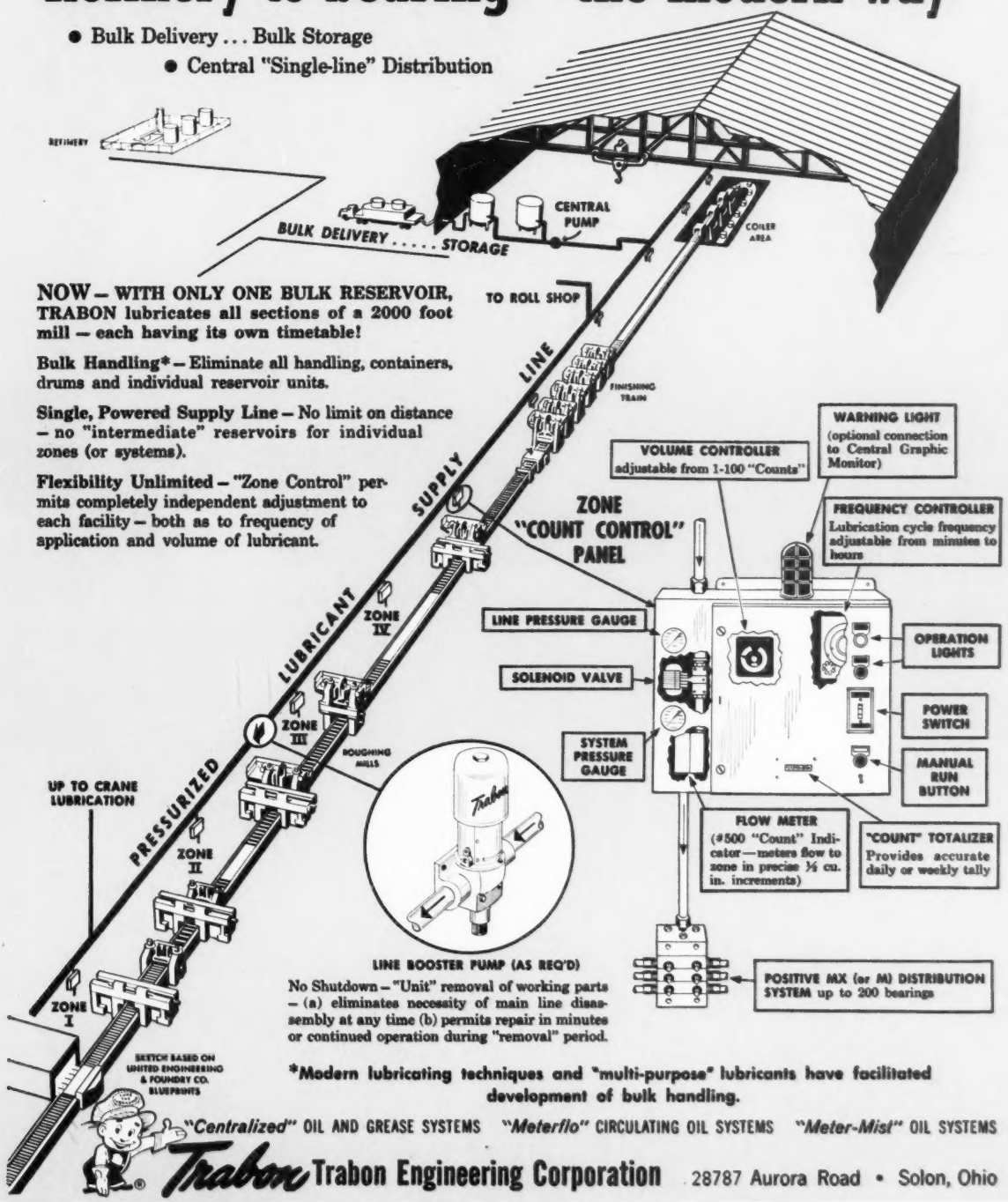
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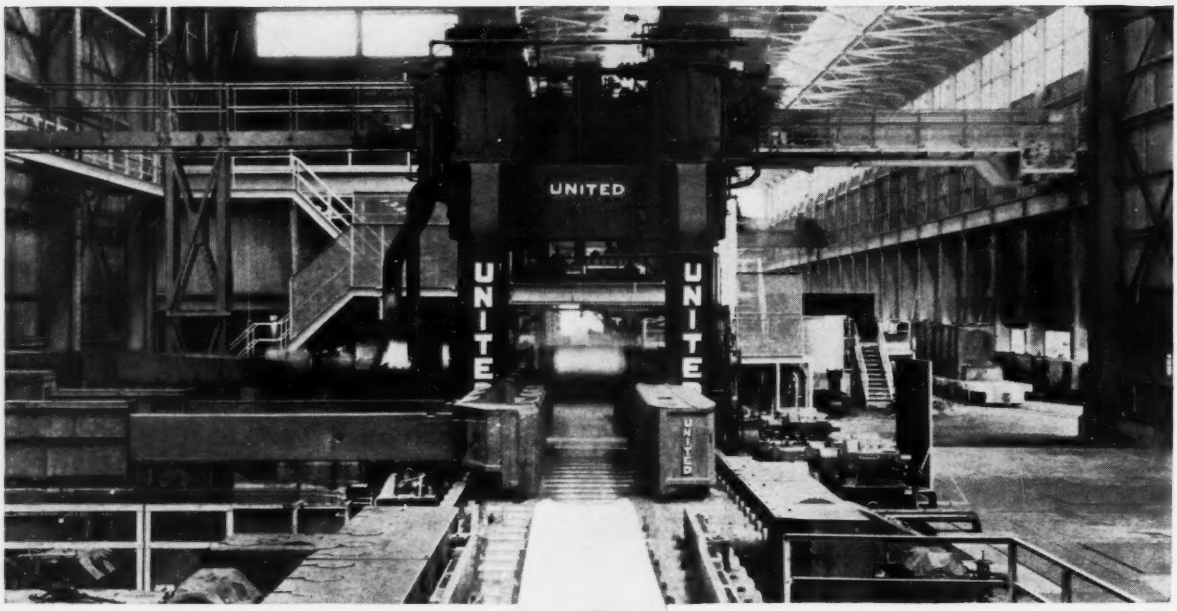

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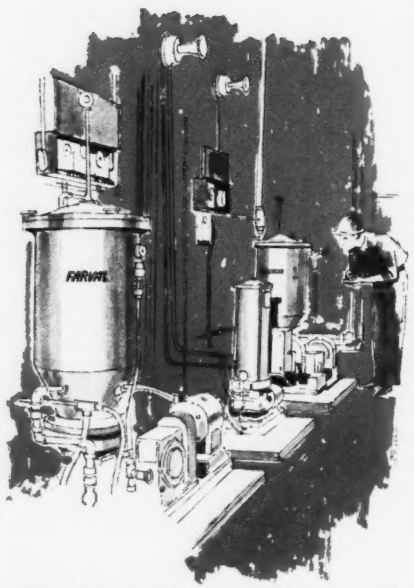

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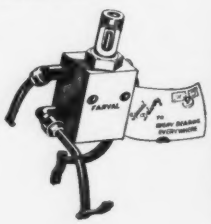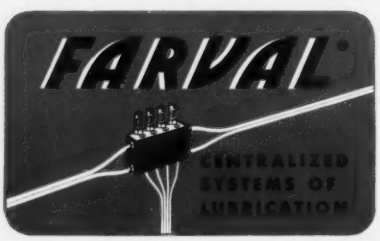
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